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MACHINE DESIGN

PART I.

FASTENINGS

BY

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PREFACE.

THE main purpose of this book is to present, in compact form for the use of the student and designer, modern American data from the best practice in the branch of Machine Design to which the work refers. The theoretical treatment of the subject has also been given fully; but this has been done for completeness only, since that field has been covered exhaustively by able writers.

Scientific analysis and the records of practice are both essential to success in the design of machine members, but neither alone is trustworthy. The former predicts only those stresses which prevail under normal conditions and ignores the overload, the rough handling, or the slight accident which the machine may meet and against which it should not fail. Practical data, on the other hand, show only the proportions which constructors have given in specific cases of stress and service and empirical formulæ founded upon them may give results wide of the mark, if the inherent limitations of these formulæ be exceeded. The problem of design is one whose many elements vary continually in number, character, and magnitude, and, for its solution, theoretical analysis, precedent, and the ripened judgment of the designer are required.

Elsewhere acknowledgment has been made of the courtesy of the many officials and companies who have furnished information. The author's thanks are due especially to Rear Admiral George W. Melville, Engineer-in-Chief, U. S. Navy; Professor Philip R. Alger, U. S. Navy; Professor J. Irvin Chaffee; Leo Morgan, Esq.; J. M. Allen, Esq., President the Hartford Steam Boiler Inspection and Insurance Company; C. C. Schneider, Esq., Vice President the American Bridge Company; Messrs. William Sellers and Company; the Baldwin Locomotive Works; and the Newport News Shipbuilding and Dry Dock Company. The author desires also to express his deep indebtedness to Stevenson Taylor, Esq., President of Webb's Academy and Vice President of the W. and A. Fletcher Co., whose examination of, and additions to, the text have added materially to the value of this work.

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MACHINE DESIGN.

CHAPTER I.

SHRINKAGE AND PRESSURE JOINTS.

Rigid connections of this character between members of a machine or structure are of frequent application. The inner member of the pair to be united is made cylindrical or slightly conical in form; the corresponding portion of the outer member is bored so that it is of the same shape, but less in diameter throughout. When, therefore, the latter is made to encircle the former, the resulting radial pressure, acting at the contact-surfaces, produces a frictional resistance to relative motion of the parts. In a *Shrinkage Fit* or *joint*, the outer member is expanded by heating, slipped in place, and held therein by the subsequent contraction in cooling. In a *Pressure* ("*Press*" or "*Forced*") *Fit*, the parts are driven together by hydraulic pressure. Joints of the latter type are, as a rule, restricted to members of moderate size — crank-pins, cranks, and the wheels and axles of engines and cars being familiar examples. The shrinkage fit is applied, not only in the union of large members in which maximum resistance to relative motion is desired, as in the crank-shafts of engines of high power; but, as well, in modern ordnance, where results of extreme accuracy are essential in order to obtain the desired inward pressure required to withstand the outward force of the gases generated in the powder-chamber.

1. General Formulæ.

The final diameter of a joint made by shrinkage or pressure is intermediate between those of the parts before union, *i. e.*, the inner member has been compressed and the outer expanded. These changes and the elasticity of the metal produce a radial compressive stress acting upon both members at the contact-surfaces and a consequent circumferential stress or "hoop-tension"

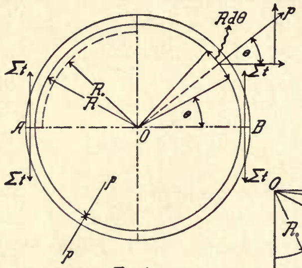


Fig. 1.

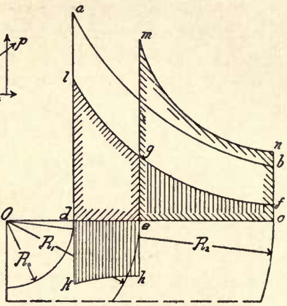


Fig. 5a.

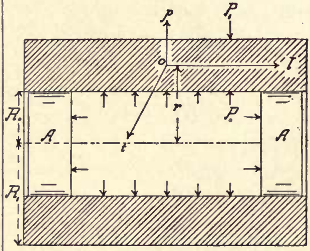


Fig. 2.

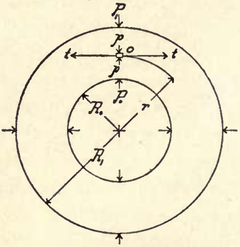


Fig. 3.

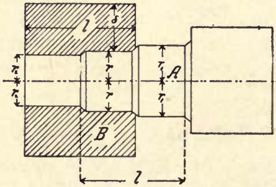


Fig. 4.

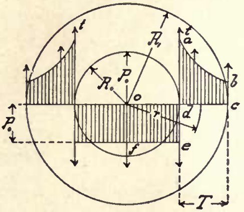


Fig. 5.

A. C. C.

within the outer member. The latter stress is a maximum at the joint and decreases rapidly toward the exterior.

1. THIN BANDS.—When the outer member is thin, as a band or tire, and the inner is, relatively, of large diameter, the compression of the latter is so small as, frequently, to be negligible in practice. The stress of the shrinkage or forcing may then be considered as expended wholly in the expansion of the band. Assume then, as in Fig. 1, an incompressible hub upon which is shrunk such a band, the stress upon the latter being within the elastic limit.

Let:

$R_0 =$ original radius of interior of band;
$R =$ radius of hub;
$t =$ tensile unit stress within band;
$e_t =$ unit elongation due to t;
$E =$ modulus of elasticity of band-metal $= \dfrac{t}{e_t}$;
$p =$ unit radial pressure;
$b =$ breadth (axial) of band;
$T =$ thickness (radial) of band, expanded;
$f =$ coefficient of friction.

Then:

$$\textit{Increase in length, interior of band} = 2\pi(R - R_0);$$

$$\textit{Original length, interior of band} = 2\pi R_0;$$

$$\textit{Elongation per unit of length} = e_t = \frac{R - R_0}{R_0};$$

$$\textit{Unit tensile stress} = t = Ee_t = E \cdot \frac{R - R_0}{R_0}. \qquad (1)$$

This tensile stress, t, acts throughout the band, tending to resist rupture of the latter on any diametral plane, as A-B. The total resistance opposed thus at A and $B =$

$$2(b \times T \times t). \qquad (2)$$

The unit radial pressure, p, acts outward, equally at all points upon the band. The latter is, therefore, virtually in the condition of a thin cylinder, of length b and thickness T, subjected to internal fluid pressure. In Fig. 1 the vertical component of the pressure p is that which tends to part the band on the horizontal

plane A-B. For an elementary strip of the band, of length $Rd\theta$, and of breadth b, we have:

Radial force on elementary strip $= Rd\theta \times b \times p$;

Parting force, elementary, on plane, $A\text{-}B = Rd\theta \times b \times p \sin\theta$;

$$\textit{Parting force, total, on band} = bpR \int_{\pi}^{0} \sin\theta d\theta = 2bpR. \quad (3)$$

Equating (2) and (3):

$$p = \frac{Tt}{R} = E \cdot \frac{(R - R_0)T}{RR_0}. \quad (4)$$

The resistance to movement at the contact-surface is equal to the product of the area of that surface, the radial pressure, and the coefficient of friction, *i. e.:*

$$\textit{Resistance to slip} = E \cdot \frac{R - R_0}{R_0} \cdot 2\pi bTf. \quad (5)$$

2. Thick Cylinders.—The method, as above, disregards the compression of the inner member, assumes the stress of forcing or shrinkage as expended wholly in expanding the band, and considers the unit-stress within the latter as uniform throughout the cross-section. The inner member cannot be incompressible and, therefore, the circumferential stress given by (1) is greater than that which would exist. The method is hence applicable only within the limits noted. In an outer member whose walls are relatively thick, the stresses at various radial distances differ widely in intensity; and, for the determination of their magnitude, recourse must be had to the complex formulæ deduced for the investigation of thick, hollow cylinders, subjected to internal fluid pressure. Of such formulæ, those founded on the method of Lamé* give, without the assumptions of Barlow or Brix, the character and intensity of the various stresses at any point within the cylinder walls.

Thus, consider, as in Figs. 2 and 3, a horizontal hollow cylinder, open at the ends, the latter being faced off in a plane normal the axis. Let this cylinder be filled with fluid, which is forced inward by two expanding plungers A, A, the result being the production of a fluid pressure upon the internal surface of the wall. From the construction and operation it is clear that, as the ends are free, the cylinder will remain a cylinder under stress;

* Rankine, "Applied Mechanics," 1869, p. 290. Burr, "Elasticity and Resistance," etc., 1897, p. 36. Cotterill, "Applied Mechanics," 1895, p. 408.

that a transverse section, taken normal to the axis when at rest, remains thus normal under stress; and that, on such a section, the resultant longitudinal stress is zero, both over the whole area and at every point thereof. Assume that the material is isotropic and that no stress, at any point, exceeds the elastic limit.

Consider any point O within the cylinder wall. Let:

R_0 and R_1 = inner and outer radii of cylinder;
P_0 and P_1 = inner and outer pressures upon cylinder;
t = circumferential stress at point O;
p = radial pressure at point O;
l = longitudinal stress = zero at point O;
r = radius of point O.

Then, from the deduction in § 10:

$$\left.\begin{aligned} t &= \frac{P_0R_0^2 - P_1R_1^2}{R_1^2 - R_0^2} + \frac{R_0^2R_1^2(P_0 - P_1)}{R_1^2 - R_0^2}\cdot\frac{1}{r^2} \\ p &= -\frac{P_0R_0^2 - P_1R_1^2}{R_1^2 - R_0^2} + \frac{R_0^2R_1^2(P_0 - P_1)}{R_1^2 - R_0^2}\cdot\frac{1}{r^2} \end{aligned}\right\} \qquad (23)$$

It will be observed that the circumferential stress t varies inversely as r^2 and is therefore a maximum at the cylinder-bore. This condition prescribes the useful limit of thickness for cylinders which are not under exterior compression. No such cylinder can be made sufficiently thick to withstand an internal pressure per sq. in. greater than the ultimate tensile strength per sq. in. of the metal, as is shown by equation (19). Since the working pressure of modern ordnance exceeds considerably the elastic limit in tension of the material used, the necessity for the "built-up" system is apparent. With regard to formulæ (23), it will be observed also that t may be either tensile or compressive, as the relations of the radii and pressures determine; that p is always compressive; and that both p and t are "apparent" and not "true" stresses, since the factor of lateral contraction has not been introduced with respect to them. Considering this factor:

$$\textit{True Circumferential Stress} = t - \tfrac{1}{3}l - (-\tfrac{1}{3}p). \qquad (6)$$

In a gun, the layer in which the breech-plug houses is under a direct longitudinal stress l, arising from the pressure upon the plug. This stress is a maximum at the face of the plug and diminishes rapidly toward the muzzle. If the apparent values of t, l and p

be substituted in (6), the working equation for true circumferential stress will be obtained, which equation is Clavarino's principal formula for the investigation and design of guns.*

3. THICK HUBS.—Professor Reuleaux, in *The Constructor*,† gives, largely without deduction, certain working formulæ, based upon those of Lamé as above, which are especially applicable to the shrinkage or cold forcing of large machine members, such as cranks and wheel-hubs. Thus, consider Fig. 4, which represents a shaft or pin A, forced into a ring or hub B. The deduction applies, theoretically, to either shrinkage or forcing. The notation is:

S_1 = radial compressive stress at r;
S_2 = circumferential tensile stress at r;
p = unit radial pressure upon contact-surfaces $= S_1$;
E_1 = modulus of elasticity, inner member;
E_2 = modulus of elasticity, outer member;
r_1 = radius of pin before forcing;
r_2 = radius of hole before forcing;
r = radius of fit;
l = length of fit;
δ = thickness, outer member, after forcing;

$$\mu = 1 + \frac{\delta}{r}; \quad \psi = \frac{r_1 - r_2}{r_2}; \quad \rho = \frac{S_1}{S_2};$$

Q = maximum forcing pressure required;
f = coefficient of friction = 0.2.

Under the conditions shown in Fig. 4, the notation of equation (23) giving the value of t, when translated into that of *The Constructor* should be changed thus:

t to S_2; P_0 to S_1; P_1 to zero; R_0 to r; R_1 to $r + \delta$; r remains as r.

(*a*) *Stresses and Allowances.*—Transforming the equation for t, in accordance with the above:

$$S_2 = S_1 \left[\frac{\left(1 + \frac{\delta}{r}\right)^2 + 1}{\left(1 + \frac{\delta}{r}\right)^2 - 1} \right] \qquad (7)$$

$$= S_1 \cdot \frac{\mu^2 + 1}{\mu^2 - 1}. \qquad (8)$$

* Merriman, "Mechanics of Materials," 1899, p. 318.
† Suplee's Translation, 1895, pp. 17, 18, 45–47.

In Fig. 4:

Unit deformation (strain), inner member $= \dfrac{r_1 - r}{r_1}$;

Unit deformation (strain), outer member $= \dfrac{r - r_2}{r_2}$.

From the definition of the modulus of elasticity:

$$\frac{S_1}{E_1} = \frac{r_1 - r}{r_1} \quad \text{and} \quad \frac{S_2}{E_2} = \frac{r - r_2}{r_2}. \tag{9}$$

Adding:

$$r_1 \frac{S_1}{E_1} + r_2 \frac{S_2}{E_2} = r_1 - r_2. \tag{10}$$

Whence:

$$\frac{r_1}{r_2} = \frac{E_1}{E_2} \cdot \frac{S_2 + E_2}{E_1 - S_1}. \tag{11}$$

By definition and from (10):

$$\psi = \frac{r_1 - r_2}{r_2} = \frac{r_1}{r_2} \cdot \frac{S_1}{E_1} + \frac{S_2}{E_2}. \tag{12}$$

From (11) and (12):

$$\psi = \frac{\dfrac{S_1}{E_1} + \dfrac{S_2}{E_2}}{1 - \dfrac{S_1}{E_1}}. \tag{36, C)*$$

S_1 and S_2 are mutually dependent, their relation being expressed by (7) and (8). In view of this and by definition:

$$S_1 = S_2 \rho. \tag{64, C}$$

From (36, C) and (64, C):

$$\psi = \frac{\dfrac{S_1}{E_1} + \dfrac{S_1}{\rho E_2}}{1 - \dfrac{S_1}{E_1}} = \frac{\dfrac{S_2}{E_2} + \dfrac{S_2 \rho}{E_1}}{1 - \dfrac{S_2 \rho}{E_1}}. \tag{37, C}$$

The second term of each denominator is so small as to be negligible. Hence:

$$\psi = \frac{S_1}{E_1} + \frac{S_1}{\rho E_2} = \frac{S_2}{E_2} + \frac{S_2 \rho}{E_1}. \tag{38, C}$$

---

* For convenience of reference, numbered formulæ from *The Constructor* are given the same numeral, with "C" added.

If the value of the ratio $\frac{\delta}{r}$, be assumed or known, it may be substituted in (7), thus giving that of the ratio, $\frac{S_1}{S_2} = \rho$, *i. e.:*

If:

$$\frac{\delta}{r} = 0.500,\ 1.000,\ 1.500,\ 2.000,\ 3.000\ ;$$

then

$$\rho = 0.385,\ 0.600,\ 0.724,\ 0.800,\ 0.882.$$

Since E_1, E_2 and the allowable value of S_2 are known quantities, the values of ψ and S_1 may be found from (38, C) by substituting the value of ρ.

(*b*) *Forcing Pressure.*—The force necessary to press a cylindrical pin into a hole by *continuous* motion may be taken as nearly proportional to the rate of progress, since that force must overcome a resistance which is largely due to sliding friction, and the latter depends upon the unit pressure on, and the area of, the surfaces in contact. The force will be a maximum just as the pin reaches the end of the hole. From Fig. 4 we have:

$$\textit{Maximum Forcing Pressure} = Q = 2\pi r \times l \times S_1 \times f. \qquad (62,\ C)$$

$$\textit{Radial Pressure} = S_1 = p = \frac{Q}{2\pi rlf}. \qquad (63,\ C)$$

(*c*) *Resistance to Slip*, either axial or rotary, is given by the value of Q in (62, C).

(*d*) *The Thickness of Hub* required to withstand the bursting pressure corresponding with the slip resistance Q, as above, may be found by combining (62, 64, C). Thus:

$$Q = 2\pi rlfS_2 \times \rho, \qquad (13)$$

in which Q is given in terms of the circumferential stress at the contact-surface. From (7), (13) and (64, C):

$$Q = 2\pi rlfS_2 \cdot \frac{\left(1 + \frac{\delta}{r}\right)^2 - 1}{\left(1 + \frac{\delta}{r}\right)^2 + 1};$$

whence $$\left(1 + \frac{\delta}{r}\right)^2 = \frac{2\pi rlfS_2 + Q}{2\pi rlfS_2 - Q};$$

and $$\frac{\textit{Thickness}}{\textit{Radius}} = \frac{\delta}{r} = \sqrt{\frac{2\pi rlfS_2 + Q}{2\pi rlfS_2 - Q}} - 1, \qquad (66,\ C)$$

from which the required thickness δ may be found.

(*e*) *Slip-resistance vs. Rotating Force.*—In (66, C) Q is the resistance opposed by the fit to slipping at the contact-surfaces; its leverage at the latter is r. Assuming the hub to be a part of a wheel or crank of effective radius R, and the external, rotating force at that radius to be P, we have, as the moment of the latter $P \times R$,

$$\therefore Q \times r \geqq P \times R. \qquad (14)$$

(*f*) *Coefficients of Friction in Forcing and Slip.*—Assuming that the resistance is wholly frictional, it is apparent that, for continuous forcing, the coefficient of friction for motion should be used. Slipping of the hub, however, must occur always from a state of relative rest of the members. Therefore in (13) and (66, C), the coefficient for rest applies. With the high radial pressures which prevail, there is a marked difference between the two coefficients.

2. Proportions of the Joint.

Economy of material prescribes that S_2 shall be the maximum permissible tensile stress. For any given fit, S_2, E_1 and E_2 are therefore constants, while the radius r is fixed by other considerations and the length l is known approximately or accurately. The total grip Q required would determine by (66, C) the value of the thickness δ, if the coefficient f were known; but experiments indicate that the value of this coefficient, as given ordinarily for the friction of motion between the clean metallic surfaces considered, is not a safe measure of the resistance of shrinkage and pressure fits, the latter especially. Such investigation, however, with regard to the value of f in these fits, has been limited. In determining δ, therefore, there should be used, preferably, formulæ which do not include this coefficient.

From (38, C) we have:

$$\frac{\textit{Total allowance}}{\textit{Diameter}} = S_2\left(\frac{1}{E_2} + \frac{\rho}{E_1}\right). \qquad (15)$$

In the right-hand member all quantities are known except ρ. From (7) and (64, C) it will be seen that, with increased thickness, ρ becomes larger. Therefore, if, with the same diameter and metals, the hub be made thicker, the total allowance, the radial pressure, and the grip per unit of surface may be increased.

Again, consider two hubs, one of steel, the other of cast iron, both on the same steel shaft, with ρ and, therefore, δ the same in each case. In the former, as compared with the latter, the circumferential stress S_2, the radial stress S_1, and the unit grip pressure may be larger and the allowance may be increased, although not proportionately. Therefore, to obtain the same grip in both cases there should be, as shown by (15), a decrease in the value of δ, ρ, and the allowance with the steel hub.

1. Allowance.—With regard to the relative values of shrinkage and forcing in producing grip, the meager experiments available indicate that, with equal allowances, fits of the former type are the more effective in resisting both torsional and axial stresses. This permits, apparently, for the same unit grip, a decreased allowance in shrinkage. The differences in grip lie, doubtless, in the methods of making the two joints. In shrinkage, there is, in cooling, simultaneous contact over the entire area of clean metallic surfaces, without relative axial movement of the latter except that due to contraction, while, in a pressure fit, surfaces lubricated originally to a greater or less extent, are not only abraded, but the passage of the inner member produces a longitudinal stress within the inner layers of the hub.

If, in (15), the quantities in the right-hand member be kept constant, there will be, for the same radial pressure and grip, a uniform allowance per inch of diameter for shrinkage or forcing. This uniformity, while by no means universal, is the practice of many large companies, a frequent allowance for steel being one one-thousandth of an inch (0.001 in.) per inch of diameter. Since the value of ρ depends upon that of the thickness, there must be also with increasing diameter a proportionate growth in thickness. When, as in Table IV., there is a decreasing unit-allowance with increased diameter, there will be lessened grip, which reduction must be met by an augmented length of hub. In any case, with diameters of 2 inches and upward, keys should be fitted between the shaft and hub as an assurance against slip.

2. Length.—Let $P \times R =$ driving moment, $\frac{J}{C} =$ polar modulus of section, $S_s =$ maximum shearing stress. Then, for a solid, cylindrical shaft of diameter, d:

$$P \times R = S_s \times \frac{J}{C} = S_s \times \frac{\pi d^3}{16}. \qquad (16)$$

From (62, C):

$$Q \times r = \pi d S_1 f l \times \frac{d}{2}. \tag{17}$$

Taking S_s and S_1 as constant and equating (16) and (17):

$$l = Kd, \tag{18}$$

where K is a constant. Therefore, with a constant radial stress, the hub-length should vary as the diameter, in order to make the grip equal to the full strength of the shaft in torsion.

3. THICKNESS.—Let Fig. 5 represent the transverse section of a closed, hollow cylinder (of inner and outer radii R_0 and R_1), initially unstressed but subjected to the internal radial pressure P_0. For these conditions, equation (23) for the stress t at radius R_0 becomes:

$$\frac{R_1}{R_0} = \sqrt{\frac{t + P_0}{t - P_0}}, \tag{19}$$

from which it appears that, if $t = P_0 =$ ultimate tensile strength, R_1 becomes infinite, *i. e.*, no thickness whatever will prevent rupture. Further, from (64, C), $P_0 = t \times \rho$, and, as ρ, in an initially unstressed cylinder, is always less than unity, the ultimate tensile stress t, as above, will be reached before P_0 attains the same intensity.

Again, for one side:

$$\textit{Area of Load Diagram O-d-e-f} = P_0 R_0;$$

$$\textit{Area of Resistance Diagram, a-b-c-d} = \int_{R_0}^{R_1} t\,dr = P_0 R_0,$$

in which $r =$ radius of any point within the wall and $t =$ tensile stress at that point, as given by (23). It is apparent, therefore, that, for any given values of P_0, R_0, and the ultimate tensile strength, there is but one value of R_1 which will satisfy the equality of the areas, as above, which value may be found from (19) by taking t at, or within, the elastic limit, making $P_0 < t$, and solving for R_1. With regard only to adequate strength, no useful purpose will be served by increasing the value of R_1 thus obtained. Finally, by substituting $\rho = \frac{P_0}{t}$ and $S_1 = P_0$ in (38, C), there will be obtained the total allowance for the prescribed diameter.

4. Form.—With regard to the form of the contact-surfaces, a slightly tapering hole and corresponding inner member have advantages over the plain cylindrical shape, in that, with the latter, the entrance of the hole must withstand the strain of abrading and compressing the pin or shaft throughout the length of the fit. The tapered member, on the contrary, enters without contact for a considerable distance and is thus well guided; the compression, upon engagement, is distributed over a greater area; the parts are separated readily when a renewal of the fit is desired; and the drawings may be marked: "Fit pin — inches from the end of the hole," which is the most trustworthy way of measuring the allowance. The disadvantage of this form lies in the difficulty of securing, with the accuracy required, the same taper in both members.

3. Metals.

From (9) it will be seen that the radial stress of the inner member and the circumferential stress within the outer, depend directly upon the modulus of elasticity E of each material so stressed. This follows since E is a measure of the stiffness of a metal, *i. e.*, the stiffer the latter, the less will be the deformation (strain) under a given stress and the larger the modulus. The following are general values:

Elastic Limit.

	Cast Iron.	*Wrought Iron.*	*Steel.*
Tension	6,000	25,000	50,000
Compression	20,000	25,000	50,000

Modulus of Elasticity.

	Cast Iron.	*Wrought Iron.*	*Steel.*
Tension	15,000,000	25,000,000	30,000,000
Compression	15,000,000	25,000,000	30,000,000

The circumferential stress of the outer member is the important element, especially when that member is of cast iron, a metal which has, in tension, a very low elastic limit, as compared with that, in compression, of the steel or wrought iron of the inner member. Cast iron is also, in tension especially, a very uncertain metal, owing to differences in composition, in the size and form of the

casting, and in the intensity of the original shrinkage strains. Professor J. B. Johnson gives E for cast iron as varying from —

"10,000,000 to 30,000,000; but, for ordinary foundry iron, it may be taken at from 12,000,000 to 15,000,000. * * * The modulus of cast iron is approximately the same in tension, compression and cross-bending." *

Professor Burr, in commenting upon certain tensile tests of cast iron, says:

"The metal is seen to be very irregular and unreliable in its elastic behavior. A large portion of the material can scarcely be said to have an elastic limit, although no apparent permanent set takes place under a considerable intensity of stress. In other words, although perhaps all tested specimens resume their original shape and dimensions for small intensities of stress, yet the ratio between stress and strain is seldom constant for essentially any range of stress."†

4. Forcing Pressures.

The pressure required, at any given time during the process, of making the joint, depends, approximately, upon the radial stress, the character and area of the surfaces in contact, and the coefficient of friction.

1. CHARACTER OF SURFACES. — This will vary with different metals and with the standard of workmanship. If the surfaces are smooth but not accurately of the same form, the radial and forcing pressures will be irregular in intensity. With rough surfaces the frictional resistance will be increased; and, in extreme cases, longitudinal cutting, uneven bearing, and lessened grip may follow.

2. COEFFICIENTS OF FRICTION. — In a pressure fit there is not only surface abrasion but the material of the outer member must be forced aside by the forward part of the advancing inner member; and, if the elastic limit of the softer metal be exceeded, some flow of the latter occurs. The resistance is not, therefore, purely frictional and the usual coefficients of friction do not give an accurate measure of its amount. In discussing shrinkage and pressure fits, Reuleaux takes $f = 0.2$ which is the value used by Weisbach for the usual metals in a dry state. The results of experiments presented in Table I. show, as a rule, much lower values of f than that quoted above. On the other hand, Rennie, from experiments upon solids usually unlubricated, gives, for pressures

* "Materials of Construction," 1898, p. 476.

† "Elasticity and Resistance of Materials of Engineering," 1897, p. 279.

per sq. in. ranging from $186\frac{2}{3}$ to 560 lbs., results, for the coefficient of rest, as follows: *

Wrought iron on wrought iron, $f = 0.25$ to 0.41;
Wrought iron on cast iron, $f = 0.28$ to 0.37;
Steel on cast iron, $f = 0.30$ to 0.36.

Abrasion occurred in the first case at 672 lbs. pressure; and, in the latter case, at 784 lbs. Broomall † gives, for static friction, as above:

Cast iron on cast iron, dry, $f = 0.3114$;
Steel on cast iron, dry, $f = 0.2303$;
Steel on steel, dry, $f = 0.4408$.

Since the value of the coefficient is affected by conditions as to motion and rest, temperature, lubrication, and speed of rubbing, reported results vary considerably. Both shrinkage and forced fits have higher radial pressures than those which prevail in the usual friction tests; the resistance in forming a pressure fit is not purely frictional; the force required to break such a joint may be less than that of making, if the elastic limit has been exceeded; and pressure fits may be lubricated only to the extent of wiping the surface with oiled waste, although a lubricant of white-lead and oil, mixed to the consistency of paint, is frequently used to prevent cutting. In view of these conditions the application to these joints of the usual coefficients for unlubricated metals, is inadvisable.

5. Shrinkage Temperatures.

Let $e =$ unit diametral or circumferential deformation; $a =$ coefficient of linear expansion for a change of one degree F.; $t =$ number of degrees of change. Assume an outer member of steel with an allowance of 0.001 in. per inch of diameter of fit. Then (Fig. 4):

$$e = \frac{r_1 - r_2}{r_2} = a \times t; \quad t = \frac{e}{a}. \tag{20}$$

Substituting:

$$t = \frac{0.001}{0.0000065} = 154^\circ \text{ F.},$$

i. e., a raise in temperature of this amount would give the members the same diameter. The usual shrinkage-temperature of wrought iron and steel is about 600°, the increase providing for greater allowance, for clearance in assembling, or for both. The value of a for cast iron is 0.0000062 per degree F.

* Thurston, "Friction and Lost Work," 1898, p. 215.
† Lineham, "Mechanical Engineering," 1898, p. 868.

6. Shrinkage vs. Pressure Fits.

Table I. gives the results of comparative tests made under the supervision of Professor Wilmore * upon cast-iron discs which were either forced or shrunk upon steel spindles, the latter being pulled from the discs in the "tension" tests or twisted in the holes in measuring the grip in torsion.

TABLE I.

No.	Fit.	Test.	Q	ψ	S_1	S_2	f
1	*P*	Tension	1,000	0.001	9,700	10,116	0.033
2	*S*	"	5,320	"	"	"	0.170
3	*S*	"	5,820	"	"	"	0.190
4	*S*	Torsion	2,200	"	"	"	0.072
5	*P*	Tension	2,150	0.0015	14,516	15,275	0.047
6	*P*	Torsion	2,200	"	"	"	0.048
7	*P*	"	2,800	"	"	"	0.061
9	*S*	"	9,800	"	"	"	0.210
10	*P*	Tension	2,570	0.002	19,355	20,366	0.042
11	*S*	"	7,500	"	"	"	0.120
12	*S*	"	8,100	"	"	"	0.130
13	*P*	Torsion	4,200	"	"	"	0.069
14	*P*	Tension	4,000	0.0025	24,194	25,458	0.053
15	*S*	"	9,340	"	"	"	0.120
16	*S*	"	9,710	"	"	"	0.130
17	*P*	Torsion	4,600	"	"	"	0.061
18	*S*	"	13,800	"	"	"	0.190
19	*S*	"	17.000	0.003	29,000	30,550	0.190

The discs were 6 in. in diameter and 1 in. thick, with, on one side, a boss 2 in. in diameter, projecting $\frac{1}{4}$ in., giving a bore $1\frac{1}{4}$ in. long and 1 in. in diameter. The spindles of machinery steel were $1\frac{1}{4}$ in. in diameter, turned at the contact-surface to 1 in. plus allowance for a length of $1\frac{1}{4}$ in., which length was reduced by a taper at the extremity and a shallow groove at the top, each $\frac{1}{8}$ in. long, making the bearing surface 1 in. in length.

The number of spindles tested was 19. The diameter of the various sets differed by 5 ten-thousandths of an inch, the finished dimensions being 1.001 in., 1.0015 in., 1.002 in., 1.0025 in. and 1.003 in. The pressure fits were made without lubrication, other than that from wiping the surfaces with oiled waste. The spindles and holes were found to be in good condition after the tests. The maximum force required to move each spindle is given as Q in the table. After movement had occurred, a less force was required to continue it or begin it anew. Columns Nos. 1 to 5, inclusive,

* *American Machinist*, Feb. 16, 1899.

of the table were taken from the data of the tests; the values in the remaining columns were computed from formulæ (15), (38, C), (62, C) and (64, C).

Accuracy in calculating the intensities of the stresses S_1 and S_2, and the coefficient f, is to some extent prevented by the boss, groove, and taper described above. The approximation given should be, however, sufficiently close for service. The value of $\frac{\delta}{r}$ was made $= \frac{2.5}{0.5} = 5$, whence $\rho = 0.946$. Since both the length and diameter of the contact surface = 1 in., ψ = allowance in each case. The coefficients E_1 and E_2, were taken as 30,000,000 and 15,000,000 respectively. Shrinkage and pressure fits are marked respectively "*S*" and "*P*," in the second column of the table.

The calculated results show very low coefficients of resistance and very high circumferential stresses. Since the ultimate tensile strength of cast-iron ranges between 15,000 and 35,000 lbs. per sq. in. and the discs were of good quality, rupture of the inner layer of the bore did not occur; but the elastic limit, in the majority of the tests, was exceeded. The superiority of the shrinkage fit is marked, as is also that of both types in torsion. Excluding tests Nos. 4 and 8, the results give average ratios of strength, as follows:

Tension: Shrinkage to Pressure = 3.66;
Torsion: Shrinkage to Pressure = 3.20;
Shrinkage: Torsion to Tension = 1.50;
Pressure: Torsion to Tension = 1.30.

7. Stationary Engines: Data from Practice.

Prevailing practice, with regard to diametral allowances in shrinkage and forced fits and the pressures required for the latter, varies considerably, owing to differences in the sizes of the members, the qualities of the metals, the workmanship upon, and lubrication of, the contact-surfaces, etc. There are given below, in tabular form, through the courtesy of leading manufacturers of stationary engines and similar machinery, records of allowances as follows:

Table II., the Lane and Bodley Company; Table III., the Russell Engine Company; Table IV., a prominent stationary engine building company; Table V., the Buffalo Forge Company; Table VI., the B. F. Sturtevant Company; Table VII., summary of Tables II. to VI.

TABLE II.*

No.	Mean Diameter of Pin (in.).	Length of Fit (in.).	Mean Diameter of Hole (in.).	Total Allowance	Allowance per Inch.	Area of Fitted Surface (sq. in.).	Volume within Fitted Surface (cu. in.).	Pressure to Enter Pin (tons).	Pressure at Mid-position (tons).	Maximum Pressure (tons).
1	1.8798	6.125	1.8767	.0031	.0017	36	16.7	2	10	20
2	1.8819	6.125	1.877	.0042	.0022	36	16.7	2	15	23
3	1.8774	4.375	1.8764	.001	.00052	24.4	13.7	½	1	1
4	2.7455	4.5	2.7387	.0068	.00247	38.7	26.5	3	12	25
5	2.7465	4.5	2.7437	.0028	.001	38.7	26.5	5	12	23
6	3.261	5	3.2542	.0068	.0021	51	41.5	5	20	45
7	3.2625	5	3.2555	.007	.002	51	41.5	5	15	30
8	3.267	5	3.261	.006	.0018	51	41.5	5	15	20
9	4.2505	6	4.2402	.0103	.0024	79.8	85.1	5	22	44
10	4.2388	6.625	4.2478	.0091	.0021	78.1	93.4	12	30	60
11	4.2303	6.5	4.2224	.0079	.0019	95.8	91	10	60	125
12	5.9343	4.0625	5.9216	.0127	.0022	75.7	112.2	6	16	25
13	5.9381	4	5.9252	.0129	.0022	74.4	110.4	3	18	35
14	5.9294	4.125	5.9194	.01	.0017	76.7	113.8	5	15	25
15	6.8829	5.125	6.8697	.0132	.002	110.7	190.1	8	20	42
16	6.889	5	6.8785	.0105	.0015	108	185.9	5	22	45
17	6.8692	4.875	6.855	.0142	.0021	104.8	180.4	5	35	65
18	7.8884	5.5	7.873	.0154	.002	135.9	267.3	5	32	64
19	7.8715	6.5	7.8575	.014	.0018	160.5	315.9	5	25	50
20	7.862	5.625	7.846	.016	.002	138.2	272.8	8	40	80
21	8.924	6.125	8.905	.019	.0021	170.8	378.9	20	45	68
22	8.9	6.75	8.8848	.0152	.0017	188.4	419.9	5	47	96
23	8.878	6.5	8.8669	.0112	.0013	180.7	401	10	45	92

TABLE III.*

CAST-IRON CRANKS.

Diameter, In.	Total Allowance, In.	
	Shrinkage.	Pressure.
4 to 5	0.0045	0.0090
5 " 7½	0.0030	0.0060
7½ " 9	0.0027	0.0055
10 " 12	0.0025	0.0050
12 " 16	0.0020	0.0040
16 " 18	0.0015	0.0030

The practice of the B. F. Sturtevant Company is as follows:

(*a*) *Shaft couplings* are bored 0.003 in. less than the shaft. The forcing pressure ranges from 6 tons for a $2\frac{13}{16}$-in. shaft to 12 tons for a 5-in. shaft.

(*b*) *Crank-pins for cast-steel crank-plates* are turned 0.005 in. large. The forcing pressure ranges from 25 to 28 tons for a 5-in. pin to 10–15 tons for small pins.

(*c*) *Crank-pins for cast-iron crank-plates* are turned 0.009 in. to 0.011 in. large. The forcing pressure is as in (*b*).

(*d*) *Cast-iron Counter-balance Plates shrunk on Steel Crank-Discs.* For diameters of 9 in. to 11 in., the total allowance is 0.007 in. With increased diameters, this allowance decreases, *i. e.*, for 13-in. diameter, total allowance = 0.006 in.

* *Machinery*, May, 1897.

TABLE IV.

(A)		(B)	
Diam., Shaft, In.	Allowance. In. of Diam.	Diam., Shaft, In.	Allowance, In. of Diam.
4	0.003	12	0.001
5	0.0024	13	0.0009
6	0.002	15	0.0008
7	0.0017	17	0.0007
8	0.0015	18	0.0006
9	0.00135	19	0.00055
10	0.0013	22	0.0004
11	0.0012	23	0.00035
12	0.011	24	0.0003
13	0.001	26	0.00025
14	0.001	27	0.0002
15	0.001		
16	0.0009		
18	0 0008		
20	0.00075		

(A) Steel shaft and pin to cast-iron cranks. Average pressure required = 12.5 tons (2,000 lbs) per in. of diam.

(B) Steel shaft to cast-iron wheel hubs. Average pressure required = 10 tons (2,000 lbs.) per in. of diam.

TABLE V.

Pressure Fits.		Shrinkage Fits.	
Diam., In.	Total Allowance, In.	Diam., In.	Total Allowance, In.
1 to 2	0.001	1 to 2	0.009
2 " 3	0.002	2 " 4	0.010
3 " 5	0.003	4 " 6	1/64 = .0156
5 " 7	0.005	6 " 9	3/128 = .0234
7 " 10	0.008	9 " 12	1/32 = .0313
10 " 12	0.010	12 " 18	3/64 = .0469

From the practice of the B. F. Sturtevant Company, with regard to crank-plates and discs, we have :

TABLE VI.

Metal.	Diameter.	Allowance Per Inch.	Type.
Cast steel.	5 in.	0.00100 in.	Pressure.
" iron.	5 "	0.00200 "	"
" "	11 "	0.00064 "	Shrinkage.
" "	13 "	0.00046 "	"

In Table II. the outer member of No. 11 was a crank-disc of cast steel, which, with less allowance, required twice the maximum forcing pressure used with No. 10. In about 75 per cent. of the fits, the maximum pressure was twice that at mid-position. The allowance for shrinkage in Table II. is one-half that for pressure (§ 2, *Allowance*), and, in both types, the unit-allowance decreases with increased diameter. The latter is true also of the fits re-

TABLE VII.

SUMMARY.

	Diameter, In.	Total Allowance, In.		Members.
		Shrinkage.	Pressure.	
Table II.	1.8798		0.0031	Shaft, steel; hub, cast iron.
" "	4.2505		0.0103	" " " " "
" "	8.9000		0.0152	" " " " "
" III.	4 to 5	0.0045	0.0090	Crank, cast iron.
" "	7.5 " 9	0.0027	0.0055	" " "
" "	16 " 18	0.0015	0.0030	" " "
" IV.	4		0.0120	" " " shaft, steel.
" "	8		0.0120	" " " " "
" "	16		0.0144	" " " " "
" V.	1 " 2	0.0090	0.0010	
" "	4 " 6	0.0156		
" "	5 " 7		0.0050	
" "	9 " 12	0.0313		
" "	10 " 12		0.0100	
" VI.	5		0.0050	Shaft, steel; crank, cast steel.
" "	5		0.0100	" " " " iron.
" "	11	0.0070		Cast-iron counter-balance
" "	13	0.0060		plates on steel crank discs.

corded in Table IV., in which, further, the allowance differs with the outer member, being less for a wheel hub than for a crank, owing doubtless to a difference in the thickness of the metal surrounding the shaft. The allowance and length of hub are so proportioned that the forcing pressure per inch of diameter is about uniform throughout the range of each type. In Table V., the allowances for pressure fits are practically uniform per inch of diameter, while those for shrinkage fits decrease with increased diameter. The latter also exceed considerably the corresponding pressure-allowances. Table VI. gives double the allowance for cast iron as compared with steel and a decreasing allowance with increased diameter.

8. Marine Engines: Data from Practice.

In marine practice, shrinkage fits are used in assembling "built-up" crank-shafts and in securing the bronze casing of propeller-shafts. Pressure fits are employed occasionally with crank-shafts and frequently with smaller work. With regard to shafts, Mr. H. F. J. Porter says:

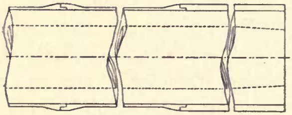
"In the built-up type, the various parts are small and can be carefully worked, and, if necessary, bored and oil-tempered. The physical properties of the metal can, therefore, be raised to the highest possible limit. The forcing or shrinking process, however, always puts a strain on the metal which will act as an initial load, approaching possibly close up to the elastic limit. In the solid type, on the contrary, a very large ingot would be required; and, as such a crooked forging cannot always be oil-

tempered with safety, the physical properties of the metal cannot usually be raised by heat-treatment. The metal, however, can be relieved of all strains by annealing; and, if properly designed, should work satisfactorily against externally applied stresses for a very long time." *

When it is possible to make the crank-shaft in sections of moderate length, interchangeable or otherwise, each section containing one or more pairs of cranks, these sections may be forged, each from a single ingot, bored and oil-tempered, thus obtaining high physical characteristics without the initial stresses due to building up.

The necessity for casing the after, or propeller, section of a marine shaft with non-corrodible material lies in the exposure of that section to the action of sea-water, both in the "stern-tube" and, sometimes, beyond the latter when the shaft extends through the water to the strut-bearing and propeller. Within the tube the bearings are of lignum vitæ and the lubricant is, as a rule, sea-water, the forward end of the tube being closed by a stuffing box. To prevent corrosion the practice, for years, has been (Fig. 6, *a*, *b*) to encase the after section of the shaft in a bronze sleeve, made in short (3-ft.) lengths, shrunk on, with lapped and recessed joints, the latter being sealed on the outside with soft solder. Since the torsion of the shaft tends to loosen the casing, the latter is secured further by pins or tap-rivets. The casing should be recessed within the propeller-hub and should make an absolutely water-tight joint with the latter. As a rule, a protecting ring of zinc is fitted also as an additional precaution against galvanic action between the casing and shaft. A less usual practice than the use of the bronze sleeve, as above, is to leave the shaft uncovered, to fit a gland at the after end of the stern-tube and to keep the latter filled with oil or tallow. In *U. S. Protected Cruisers, Nos. 20 to 22*, the diameter of the propeller-shaft is 18 in. and the casing thicknesses are 1 in. at forward and $1\frac{1}{16}$ in. at after bearing, $\frac{7}{8}$ in. at the laps (1 in. long), and $\frac{3}{8}$ in. elsewhere. The following data are given through the courtesy of leading builders of marine work.

Fig 6b.



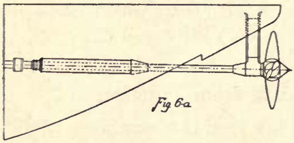


Fig 6a.

* "Fatigue of Metal," etc., *Jour. Franklin Institute*, Dec., 1897.

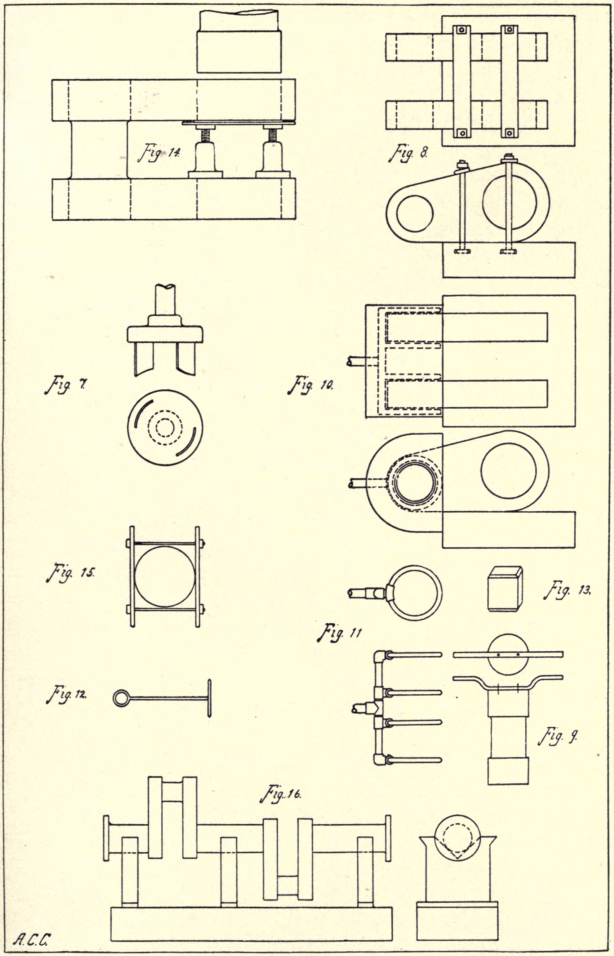


Fig. 14
Fig. 8
Fig. 7
Fig. 10.
Fig. 15.
Fig. 13.
Fig. 11
Fig. 12.
Fig. 9.
Fig. 16.
A.C.C.

1. The practice of the Midvale Steel Company, Philadelphia, Pa., is as follows:

(*a*) *Shaft Casings.*—A new stern-tube shaft for the American Liner *New York* was made recently at these works. It was 40 ft. long, 20½ in. diameter, and was cased partially with two bronze sleeves, each 8 ft. long, fitted by shrinkage, the total allowance for the latter bring 0.013 in. = 0.000634 in. per inch of diameter. To secure uniform expansion, the casing was set vertically and heated internally by gas, the latter issuing from a pipe a little longer than the sleeve, inserted within the latter, and perforated for the flow. When the bore as gauged showed sufficient expansion for a free fit, the sleeve was slipped in place, held firmly at one end, and cooled by water at the latter until contraction and grip occurred.

(*b*) *Crank-shafts.*—An allowance of 0.001 in. per inch of diameter is made for steel. The method of building up is shown in Figs. 7 to 16, inclusive. The crank-pin is finish-machined and a cross-piece (Fig. 9), for guiding it when inserted, is secured by screws at one end. The holes in the crank-webs for pin and shaft are bored in a vertical machine to within ¼ in. of finished diameter, the tool (Fig. 7) being circumferential and two-bladed. If the web is less than 7 in. thick, the cut is made from one side in one setting; otherwise, it is run half way through from each face. Then the two webs which form a pair are bolted to a portable surface-plate (Fig. 8), the latter is set on a horizontal machine, and the holes are bored to the diameter of the pin, less the shrinkage-allowance. The setting on the plate, with regard to parallelism and distance, is that required for the pin when the latter is in place.

The webs are then heated in a sheet-iron furnace (Fig. 10), provided with a burner of perforated gas-pipe (Fig. 11), sliding doors, and covered holes for occasional measurement of the bores by a gauge (Fig. 12) made to the exact diameter of the pin, the gauge being cooled in water after each test. When the expansion is sufficient for a free fit, the webs are removed from the furnace and the pin is pushed home, being guided by the cross-piece so that the key-ways come flush, the latter being ensured by a loose false key (Fig. 13) which is inserted as soon as the pin enters the web. The pin is slung from a crane-hook, the sling being shifted, if the pin is solid, when the latter has traversed one hole. If the pin is hollow, it rides on a heavy gas-pipe, passing through the bores and suspended by slings at the ends.

The webs and pin are cooled with water, the false key is taken out, and the permanent key driven home. The construction is then removed from the surface-plate and set in a horizontal machine, where the holes for the shaft are bored to the finished diameter, less shrinkage-allowance. The webs are then set with the bores vertical and one is heated as before. When the furnace is removed, a planed plate (Fig. 14) is placed under the heated web, a paper liner—which does not project into the bore—is laid between, and the plate is forced against the web by three or four screw-jacks. The shaft is then slung vertically over the bore and lowered until it meets the plate, the downward projection due to the liner being sufficient to make the end of the shaft and the face of the web flush, when cooled by water. False and permanent keys are fitted, as with the pin. While lowering, the shaft is guided by a wooden frame (Fig. 15).

The remaining portion of the shaft is then shrunk into the other web; the completed section is set in a lathe; the shaft and pin are tested for parallelism; and the centers of the shaft are drawn to correct any error. The section is then finish-machined and joined by shrinkage with others. The entire shaft is then placed in a line of V-blocks (Fig. 16), accurately set on a bed, for the final tests in calipering, parallelism of center-lines, faces of webs and couplings, and to determine whether the two latter are square with the center-lines. Any errors detected are corrected by hand-work.

2. Examples of the practice of the Union Iron Works, San Francisco, Cal., are given in Table VIII.

TABLE VIII.

Members.	Diam. Ins.	Total Allowance.		Forcing Pressure, Tons.
		Shrinkage.	Pressure.	
Steel Crank to Steel Shaft.	14	0.015625	0.00938	100 to 150
Wro't-iron Crank to Wro't-iron Shaft.	8	0.0125	0.007	80 " 100
Cast-iron Crank to (hard) Steel Shaft.	8		0.00938	80
" " " " (soft) " "	8		0.00938	20
Wheel Hub (C. I. hard) " " Length of Fit, 36-in.; Mean Dia'r.	17–63/64		0.003125	80
As above; hub of soft cast iron.			0.003125	30
Cylinder-Liner, cast iron, hard.* In Cylinder " " medium.	80		0.0219	
As above.	60		0.015625	40 to 60
" "	30		0.0125	

* Pressure fits now discontinued.

3. The practice of the New York Shipbuilding Company, Camden, N. J., is as follows:

(*a*) *Allowances.*—These, in shrinkage or pressure fits in iron or steel, are one one-thousandth of an inch (0.001 in.) per inch of diameter of fit, plus one one-thousandth of an inch (0.001 in.). Thus, on a 2-in. diameter, the allowance is 0.003 in.; on a 10-in. fit, 0.011 in., etc.

(*b*) *Form.*—With large fits, both the inner and outer members have a taper of $\frac{1}{16}$ in. to the foot, the allowances being as above, If the conditions are such that it is more convenient to ream the hole with standard parallel reamers, the inner member is tapered one half thousandth of an inch (0.0005 in.) per inch of length, unless the fit is so long that this taper would reduce the allowance at the small end to less than one half that at the other extremity of the joint.

(*c*) *Drive Fits.*—For these, the allowance is one half that for shrinkage or pressure joints.

(*d*) *Shaft-Casings.*—The allowance is one half that for a shrinkage fit on heavy work.

4. The Harlan and Hollingsworth Company, Wilmington, Del., give, in built-up shafts, a shrinkage allowance of one one-thousandth of an inch (0.001 in.) per inch of diameter; and, in shaft-casings, one half of this amount, *i. e.*, 0.0005 in.

9. Railway Work: Data from Practice.

In railway work pressure fits are used in securing wheels to axles and crank-pins to driving wheels while the tires of the latter are shrunk in place. A pair of drivers consists of the axle of wrought iron or steel, the wheel-centers of cast iron, the tires of

TABLE IX.

Diameter, Wheel Center, In.	Total Allowance, Tire, In.	
	A	B
38	0.040	0.0312 = 1/32
44	0.047	0.0469 = 3/64
50	0.053	0.0625 = 1/16
56	0.060	0.0625 = 1/16
62	0.066	0.0781 = 5/64
66	0.070	0.0781 = 5/64

steel, and the crank-pins of the latter metal. In assembling these parts, the wheel-centers are first driven on the axles and keyed.

The tires are then shrunk on, the holes bored for the crank-pins and the latter pressed in. Finally, the tires are turned to the finished size.

1. TIRES. — In 1886–7 the American Railway's Master Mechanics' Association recommended and adopted the diameters and allowances printed, through the courtesy of that Association, in the first and second columns of Table IX. These allowances have not met universal use; and, in column B the practice of a prominent road, for the same diameters, is presented. The fit is cylindrical between the wheel-center and the tire. The latter is heated usually by gas-jets set about its circumference; and, when expanded, is placed on the wheel-center and allowed to cool. Tires thus secured resist the lateral thrust and rolling action until they are worn considerably, when they may become loose and require liners or refitting.

2. WHEEL-FITS.—The joint is cylindrical. The pressure required for mounting the wheel is usually 9 to 10 tons per inch of diameter of fit; for removal, the total pressure may be 100 to 150 tons, depending on the condition of the joint as to rust, etc. The

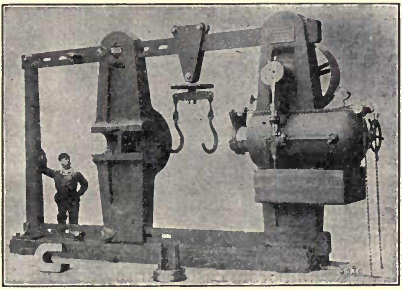


FIG. 17.

mechanism used in these operations is shown by Figs. 17 and 18, which represent the 400-ton wheel-press made by, and illustrated herein through, the courtesy of the Niles Tool Works Company, Hamilton, Ohio.

The press consists essentially of a hydraulic ram; a resistance head, or abutment, sliding on tension-bars to which it may be keyed at the required distance from the ram-head; and supporting

hooks for the axle, depending from the upper bar. The resistance-head has a central bearing for the axle, to enable the latter to lie in the line of pressure. In mounting wheels, the axle, with each wheel started on its fit, is hoisted into the hooks and resistance-head, and the ram, acting on the hub next to it, drives both wheels home. In dismounting, the resistance-head is moved nearer to the ram, the stop-block shown in the head is removed, and the axle is laid within the latter. The ram then engages the axle and forces it out of the wheel, after which the axle is reversed and the remaining wheel removed in a similar way.

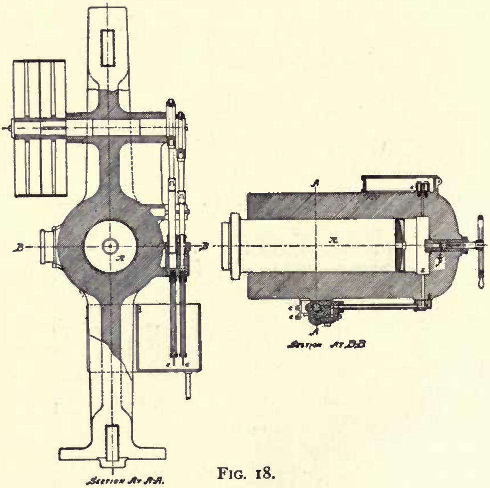


FIG. 18.

The ram R is a solid iron casting, provided, at the rear, with cupped leather packing. The cylinder is of strong and dense cast iron, lined with $\frac{1}{8}$ in. copper, the latter being spun into place and beaded over the counterbore. Water from the pumps enters at d; a release-valve f, operated by a hand-wheel, permits the fluid to escape, when desired, into the tank; a safety-valve, e, limits the pressure to 6,000 lbs. per sq. in.; and the chains and counter-weight retract the ram when the release valve is opened.

The pump is provided with two plungers, $1\frac{3}{4}$ in. and 1 in. diameter, respectively, each operated by an eccentric on the driving

shaft. The plunger chambers are separate, each being provided with suction and discharge valves. Through the suction pipe to each chamber a tripping rod, *c*, passes, which, when elevated, lifts the suction valve from its seat and thus stops the delivery from that chamber while the shaft still rotates. The rod, *c*, is connected externally to a lever and link, a support holding the latter in place when the suction-valve is operating. It will be seen that the tripping rods provide a very quick method of throwing either or both pumps out of operation—an action which is essential, since, when the wheel has reached the end of the fit, the inflow to the cylinder should cease at once.

In starting the press, the belt is shifted to the tight pulley, the trip-rods are lowered and both plungers operate until such a pressure has been obtained as the belt permits. Then the suction valve of the larger chamber is tripped and the work continues with the smaller plunger until the limit of the fit is reached, when the remaining suction valve is raised and further movement of the ram is prevented.

10. Shrinkage in Gun Construction.

The stresses to which a gun is subjected upon the explosion of the charge are: First, a radial pressure tending to split it on an axial plane; and second, a longitudinal stress acting to rupture it on a plane transverse to the axis. There must be considered also in design the radial compression of the bore — due to the shrinkages of the exterior cylinders — which, when the system is at rest, the inner layer must withstand.

To secure equal strength throughout without undue weight, the material should be so arranged that every portion does its full share in resisting the pressure from within. Fig. 5 shows the rapid reduction in stress toward the exterior of a homogeneous cylinder, the tension in the outer layer being but two fifths of that in the inner, when $R_1 = 2R_0$. This uneconomical distribution of the metal and the fact that the elastic strength of the latter is, in such cylinders, the limit of the allowable internal pressure P_0, led to the abandonment of cast guns, although some measure of compressive, reinforcing stress upon the bore may be obtained, during casting, by cooling the inner wall first, thus producing tension in the outer layers.

Maximum economy of material will be attained when the stresses throughout the walls are, at all points, upon the explosion of the charge, not only approximately equal but also the greatest permitted by the elastic strength. This condition can be approached only by placing the outer metal in a state of initial tension, the result being, when the system is at rest, a compression and reinforcement of the inner layer, the latter being given thus additional strength, since the initial compression must be overcome by the pressure of the gases before tensile stress in the fibers will be produced. In order to develop these initial stresses, the gun is built of separate concentric cylinders shrunk one upon the other, the unit diametral allowance or *relative shrinkage* of the outer cylinders being such that, while these cylinders are thus normally in tension, they have still a margin of strength, within their elastic limits, to withstand the added tensile stress upon explosion. The stress-diagrams for such a construction are shown approximately in Fig. 5, *a*, which represents a portion of a transverse section of a tube with superposed cylinder. The area, *a-b-c-d*, is the diagram of tangential stress for a single cylinder of the maximum radius and combined thickness, subjected to the internal pressure, P_0. The area, *e-g-f-c*, represents the initial tension in the outer cylinder, and its equivalent, *d-e-h-k*, the initial compression in the tube. The areas, *d-l-g-e* and *e-m-n-c*, show, respectively, the tangential stresses in the tube and cylinder when under the internal pressure, P_0. It is obvious that the latter areas are together equal to the original diagram, *a-b-c-d*, less that of initial compression, and plus that of initial tension. The possibility of reducing the stress at the bore is apparent. Since both radial and circumferential stresses change with each increment of radius, the greater the number of superposed cylinders in a given thickness, the more equable will be the disposition of stress under internal pressure. In practice (Fig. 19), the number of such cylinders is, in large guns, four, viz.: the *tube*, a single forging, the length of the bore; the *jacket*, encircling the tube from the breech-end about half way to the muzzle; two layers of *hoops*, superposed upon the jacket, the chase-hoop extending to the muzzle; and tapering and locking *bands*. With regard to the radial and circumferential stresses in a gun thus assembled, Major Rogers Birnie, U. S. A., says:

"The accepted theory of this mode of construction is to assemble the several rows of cylinders so that :

"In whatever state the system may be considered, none of the fibers of any cylinder in the structure shall be elongated or contracted beyond the elastic limits determined for such displacements by the free tests of the metal.

"With the *system at rest* this applies especially to the tube which, ordinarily, has to support alone, or without other assistance than the atmospheric pressure, the accumulated stress due to the shrinkages of all the outside cylinders. Under these circumstances, the surface of the bore undergoes the greatest change of form by compression, so that the shrinkages of the outer cylinders must be limited to retain uninjured the elastic properties of the metal at the surface of the bore of the tube. (It is, perhaps, an open question whether the compression of the bore may not, with advantage, be carried beyond this limit; but, for the purposes of theoretical discussion, we assume that it should not be.)

"With the *system in action*, that is, subjected to the maximum interior pressure which it can support with safety, the cylinders or hoops composing each layer of the structure should work together to the elastic limit of their metal. Here, again, it is the interior fibers which undergo the greatest change of form in general by circumferential extension in the outer cylinders and by radial compression in the inner cylinders. The theoretical resistance of the gun must then be limited to retain uninjured the elastic properties of the metal at the interior of any of the cylinders composing the structure. This involves the following considerations, viz.: As many of the cylinders as practicable should work together to the elastic limit of their material under extension; but, when other cylinders are endangered from radial compression of their walls, the theoretical interior pressure must be curtailed to provide against such over-compression, and the working tensions of the first-named parts will be correspondingly reduced. However, the wall of the tube (or part of the structure next to the bore) has always to support the greatest normal pressure with the system in action; hence, frequently, in this state of the system also, the theoretical resistance of the gun will be limited by the strength of the tube to resist compression, in this case radial instead of tangential, as in the other extreme state of the system." *

Major Birnie considers that the longitudinal tension developed in firing may, without noteworthy error, be neglected in deducing the equations of equilibrium, expressing the relations between the tangential and radial resistances for any state of the system.

1. SHRINKAGE FORMULÆ.—For the deduction which follows the author is indebted to Professor Philip R. Alger, U. S. N., formerly of the Bureau of Ordnance, U. S. Navy, now head of the Department of Mechanics, U. S. Naval Academy. Practically all of the guns in the U. S. Navy were assembled with shrinkages calculated by the formulæ given below.

In this deduction it is assumed:

1. That there is no longitudinal stress on any layer. This would be true only in the case of a hollow cylinder under fluid

* Ordnance Department U. S. A., "Notes on the Construction of Ordnance," No. 35.

pressure and having both ends free, and is not true for a gun; but, even with the latter, only the layer in which the breech-plug houses is under direct longitudinal stress and that stress diminishes rapidly as we go forward from the breech-plug face.

2. That a transverse section of the cylinder when at rest remains a plane normal to the axis of the cylinder when the latter is under strain—in other words, that the longitudinal strain is uniform over the whole section. This would be a natural result of the condition of free ends, but can be considered as only approximately true for a gun.

3. That the total strain, in any direction, due to all the stresses is the measure of the tendency to yield in that direction, so that the limit of safety is reached, not when the stress in any direction equals the elastic strength of the material, but when the strain in any direction equals the strain which would be caused by the direct action of a single stress equal to that elastic strength.

4. The ratio of strain, in the direction of the stress producing it to the accompanying strain at right angles to that direction, is taken to have the value 3.

(*a*) *Stresses and Strains.*—Let a hollow cylinder of radii R_0 and R_1 be under pressure P_0 from within and P_1 from without, and let T_0 and T_1 be the resulting circumferential tensions at the inner and outer surfaces. Also, let t and p be the circumferential tension and radial pressure at any point of radius r within the cylinder-wall and let e_t, e_r and e_l be the tangential, radial and longitudinal strains at the same point. Also, let E be the modulus of elasticity of the material. Then:

$$e_t = \frac{1}{E}\left(t + \frac{p}{3}\right);\quad e_r = -\frac{1}{E}\left(p + \frac{t}{3}\right);\quad e_l = -\frac{1}{E}\left(\frac{t}{3} - \frac{p}{3}\right) \qquad (21)$$

and since, by hypothesis, e_l is constant, we have

$$t - p = constant = k.$$

But

$$\int_{R_0}^{R_1} t dr = P_0R_0 - P_1R_1;$$

and, assuming $t = f'(r)$, this gives,

$$f(r)\Big]_{R_0}^{R_1} = P_0R_0 - P_1R_1;$$

whence

$$f(r) = -pr; \text{ and so, } t = f'(r) = -p - r\frac{dp}{dr}.$$

Thus, we have $t - p = k$ and $t + p = -r\frac{dp}{dr}$, whence

$$2p + k = -r\frac{dp}{dr},$$

the integration of which gives $2p + k = \frac{k_1^2}{r^2}$, where k_1 is a constant of integration. Combining with $t - p = k$, we have $t + p = \frac{k_1^2}{r^2}$.

These, then, are the fundamental equations which express the relation between circumferential tension and radial pressure at all points within the cylinder:

$$\left.\begin{aligned} t - p &= k = T_0 - P_0 = T_1 - P_1 \\ (t + p)r^2 &= k_1^2 = (T_0 + P_0)R_0^2 = (T_1 + P_1)R_1^2 \end{aligned}\right\} \quad (22)$$

Eliminating T_1 between the last parts of these equations, we have:

$$T_0 = P_0 \cdot \frac{R_1^2 + R_0^2}{R_1^2 - R_0^2} - \frac{2R_1^2 P_1}{R_1^2 - R_0^2},$$

and substituting this in the first parts of the same equations, we have, after combining:

$$\left.\begin{aligned} t &= \frac{P_0R_0^2 - P_1R_1^2}{R_1^2 - R_0^2} + \frac{R_0^2R_1^2(P_0 - P_1)}{R_1^2 - R_0^2} \cdot \frac{1}{r^2} \\ p &= -\frac{P_0R_0^2 - P_1R_1^2}{R_1^2 - R_0^2} + \frac{R_0^2R_1^2(P_0 - P_1)}{R_1^2 - R_0^2} \cdot \frac{1}{r^2} \end{aligned}\right\} \quad (23)$$

Substituting these values in the first part of (21), we have, for the tangential strains at the inner and outer surfaces, where $r = R_0$ and $r = R_1$, respectively:

$$\left.\begin{aligned} e_{T_0} &= \frac{1}{E} \cdot \frac{P_0(2R_0^2 + 4R_1^2) - 6P_1R_1^2}{3(R_1^2 - R_0^2)} \\ e_{T_1} &= \frac{1}{E} \cdot \frac{6P_0R_0^2 - P_1(4R_0^2 + 2R_1^2)}{3(R_1^2 - R_0^2)} \end{aligned}\right\} \quad (24)$$

Suppose now the pressure P_1 to be caused by a second cylinder (radii R_1 and R_2) embracing the first and itself under the external

pressure P_2. Let the circumferential tension at its inner surface be designated as T_1' (to distinguish it from T_1, the tension of the outer surface of the inner cylinder, which is under the same radial pressure P_1, but not at the same tension as the surface in contact with it) and that at its outer surface as T_2. Then, applying formula (24) to this second cylinder, we have, for the circumferential strains at the inner and outer surfaces:

$$\left.\begin{aligned} e_{T_1'} &= \frac{1}{E}\cdot\frac{P_1(2R_1^2+4R_2^2)-6P_2R_2^2}{3(R_2^2-R_1^2)} \\ e_{T_2} &= \frac{1}{E}\cdot\frac{6P_1R_1^2-P_2(4R_1^2+2R_2^2)}{3(R_2^2-R_1^2)} \end{aligned}\right\} \qquad (25)$$

Finally, assuming P_2 to be caused by a third cylinder (radii, R_2 and R_3) whose outer surface is under no pressure, we have, for the circumferential strain at its inner surface:

$$e_{T'} = \frac{1}{E}\cdot\frac{P_2(2R_2^2+4R_3^2)}{3(R_3^2-R_2^2)} \qquad (26)$$

Now let $\frac{\theta_0}{E}$, $\frac{\theta_1}{E}$, and $\frac{\theta_2}{E}$ be the values fixed for the maximum strains of the three cylinders respectively, when under the action of the system of pressure P_0, P_1 and P_2. Substituting these values for e_{T_0}, $e_{T_1'}$, and $e_{T_2'}$ in (24), (25), and (26), we have

$$\left.\begin{aligned} P_2 &= \frac{3(R_3^2-R_2^2)}{4R_3^2+2R_2^2}\cdot\theta_2 \\ P_1 &= \frac{3(R_2^2-R_1^2)\theta_1+6P_2R_2^2}{4R_2^2+2R_1^2} \\ P_0 &= \frac{3(R_1^2-R_0^2)\theta_0+6P_1R_1^2}{4R_1^2+2R_0^2} \end{aligned}\right\} \qquad (27)$$

the last of which equations gives the internal pressure which the built-up cylinder will stand, if its parts have been so assembled that the inner surface of each reaches at the same instant the condition of maximum circumferential strain assigned to it. This, of course, implies a definite shrinkage for each cylinder, which shrinkage remains to be determined.

(*b*) *Relative Shrinkages.*—Observe now that equations (24), (25) and (26) give the tangential strains resulting from the pressures P_0,

P_1, and P_2, and that if we substitute for these pressures any simultaneous changes in their values as p_0, p_1, and p_2, the same equations will give the corresponding changes of strain. But the surfaces of contact of the cylinders must contract and expand together and so the change of strain at the outer surface of each cylinder must equal that simultaneously occurring at the inner surface of the cylinder embracing it. Hence equating the second part of (24) to the first part of (25) and the second part of (25) to (26), after replacing P_0, P_1, and P_2 by p_0, p_1 and p_2, we have:

$$\left.\begin{array}{l} R_0^2(R_2^2-R_1^2)p_0-R_1^2(R_2^2-R_0^2)p_1+R_2^2(R_1^2-R_0^2)p_2=0 \\ R_1^2(R_3^2-R_2^2)p_1-R_2^2(R_3^2-R_1^2)p_2=0 \end{array}\right\} \quad (28)$$

the first of which gives the relation between simultaneously occurring changes in the pressures at the radii, R_0, R_1, and R_2, and the second, the relations between such changes at the radii, R_1 and R_2.

If, now, in the first equation of (28), we make $p_0=-P_0$ and $p_2=-P_2$, we find:

$$p_1=-\frac{R_0^2(R_2^2-R_1^2)P_0+R_2^2(R_1^2-R_0^2)P_2}{R_1^2(R_2^2-R_0^2)};$$

and this is the change of pressure at the radius R_1, which would result from the simultaneous removals of the outer cylinder which causes P_2 and of the internal pressure P_0 itself. Therefore, substituting this value of p_1 for P_1 and $-P_2$ for P_2 in the second equation of (25), we have, for the change of outer diameter of the middle cylinder, due to removing the outer cylinder and suppressing the internal pressure, the expression:

$$\frac{1}{E}\cdot\frac{(4R_0^2+2R_2^2)P_2-6R_0^2P_0}{3(R_2^2-R_0^2)}.$$

But, by hypothesis, the strain at the inner surface of the outer cylinder, before the change just referred to, was $\frac{\theta_2}{E}$, and, therefore, the relative shrinkage of the outer cylinder must have been:

$$\varphi_2=\frac{1}{E}\left[\theta_2+\frac{(4R_0^2+2R_2^2)P_2-6R_0^2P_0}{3(R_2^2-R_0^2)}\right]. \quad (29)$$

To find φ_1, the relative shrinkage of the middle cylinder, put $-P_0$ for P_0 and $-P_1$ for P_1 in the second equation of (24) which

gives, for the change in outer diameter of the inner cylinder, due to removing the outer cylinders and suppressing the internal pressure, the expression:

$$\frac{1}{E}\cdot\frac{(4R_0^2+2R_1^2)P_1-6P_0R_0^2}{3(R_1^2-R_0^2)},$$

whence

$$\varphi_1=\frac{1}{E}\left[\theta_1+\frac{(4R_0^2+2R_1^2)P_1-6R_0^2P_0}{3(R_1^2-R_0^2)}\right]. \qquad (30)$$

By the term *relative shrinkage* is meant the difference of diameter per unit length of diameter of the surfaces to be superposed, so that the actual differences of diameter are $2R_2\varphi_2$ and $2R_1\varphi_1$.

(*c*) *The Method of Procedure*, then, is to calculate P_2, P_1 and P_0 by formulæ (27) and then determine the shrinkages by formulæ (29) and (30). It may be, however, that the shrinkages thus found would cause excessive compression of the bore of the inner cylinder, when at rest; and, if so, smaller values of θ_1 and θ_2 must be used. To ascertain whether this is the case, eliminate p_2 between the parts of equation (28) which gives:

$$p_1=\frac{R_0^2(R_3^2-R_1^2)}{R_1^2(R_3^2-R_0^2)}\cdot p_0;$$

and, making $p_0=-P_0$ in this, the resulting value of p_1 is the change of pressure at the outer surface of the inner cylinder due to the suppression of P_0. Therefore, p_1+P_1 must be the pressure on that outer surface when the system is at rest; and this must not exceed

$$\frac{R_1^2-R_0^2}{2R_1^2}\cdot\theta_0,$$

since, if it does, the tangential compression of the bore will exceed θ_0.

As a matter of fact, however, experience seems to show that there is no objection to compressing the bore beyond the elastic limit of the material under tension, presumably because the elastic resistance to compression is really considerably greater than that so-called elastic limit of tension.

It is also to be noted that no account has been taken of the fact that the radial strain at the inner surface of a cylinder may, and

indeed sometimes does, exceed the tangential strain, while our formulæ assume that it is only the latter which must not exceed a fixed limit. This, too, can only be justified by the assumption that the material really has a higher limit of elasticity under compression than under tension.

In assembling U. S. naval guns with shrinkages calculated by the foregoing formulæ, θ_0, θ_1 and θ_2 were taken as the lowest elastic limit given by any specimen from the particular forging considered, excepting where the resulting compression of bore considerably exceeded θ_0, in which case θ_1 and θ_2 were somewhat reduced. The formulæ as given herein are, of course, easily extended to cover cases where there are more than three layers.

The *tangential strain* is really the change of length per unit length of the circumference and, so also, the change of length per unit length of diameter. An alternative nomenclature of the strains is as follows: Take a circle of radius r in the cylinder walls when at rest and suppose that, when the pressures act, each point of the circle moves outwardly Δr and axially Δh, then the tangential strain is $\frac{\Delta r}{r}$, the radial strain is $\frac{d\Delta r}{dr}$, and the longitudinal strain is $\frac{d\Delta h}{dh}$, these strains being what have been called e_t, e_r and e_l.

(d) *Radii.*—If only the *tangential* resistance to internal pressure is to be considered, the maximum value of P_0 will be obtained by making the radii increase in geometrical progression from that of the chamber outward, *provided* the several cylinders have the same elastic strength and the same modulus of elasticity. Thus, for the case of one cylinder superimposed upon another, make P_0, formula (27), a function of R_1 (R_0 and R_2 being constant and $\theta_1 = \theta_0$), differentiate, and make $\frac{dP_0}{dR_1} = 0$. After cancellation, we have $R_1^2 = R_0R_2$, showing that the maximum value of elastic resistance for a given total thickness of a given material occurs when the radius of the common surface is a mean proportional between the inner and outer radii. For example, with the 6-inch gun of 4-inch chamber-radius and 8-inch thickness of chamber-wall, the maximum resistance against tangential bursting stress would be secured by making $R_0 = 4$-inch; $R_1 = 4\sqrt[3]{3}$; $R_2 = 4\sqrt[3]{9}$; and $R_3 = 4\sqrt[3]{27} = 12$.

In practice, however, other considerations than tangential stress prevent complete conformity with theory. In the first place, it is necessary to make that layer which takes the longitudinal strain of sufficient cross-section. In United States guns, the breech-block houses in the jacket or second layer and the area $\pi(R_2^2 - R_1^2)$ must be adequate, being, in naval guns, about three times that of the rear end of the chamber, so that the longitudinal stress on the jacket, if uniformly distributed, is one third of the chamber pressures. In French guns, the breech-block usually houses in the tube or inner layer, thus making R_1 much greater than is necessary for resistance to the maximum tangential stress.

Again, the tube thickness over the enlarged chamber should not be too small to prevent lining the bore with a thin tube, after the erosion of the powder gases has cut away the rifling and rendered the gun inaccurate. Finally, the necessity for keeping down weight, which prescribes a decreasing exterior diameter to correspond with the diminishing pressure toward the muzzle, together with the need for avoiding sudden or great changes of section in the various forgings, sometimes dictates dimensions not otherwise desirable.

2. Gun Construction.—The 16-inch Breech-loading Rifle (Type, Model 1895), completed—except as to the final boring, rifling, and the hoops engaging the mount—during the year 1900 by the Ordnance Department, U. S. A., at the Watervliet Arsenal, N. Y., is not only the most powerful gun yet built, but is also the largest construction ever assembled by shrinkage. The general data * are as follows:

Weight of gun 126 tons (252,000 lbs.), of armor-piercing projectile, 2,400 lbs., of powder-charge (smokeless), 576 lbs.; powder-pressure, 37,000 to 38,000 pounds per sq. in.; muzzle-velocity, 2,300 ft. per second; muzzle-energy, 88,000 ft.-tons; penetration in steel at muzzle (De Marre's formula, normal impact), 42.3 in.; range, 20,978 miles; height of trajectory, 30,516 ft. (about 5¾ miles); length of projectile, 5 ft. 4 in.; cost per round, powder and shot, $1,000.

(*a*) *Description*.—The gun is shown in section in Fig. 19. Its total length is 590.9 in.; external diameter at rear, 60 in., at muzzle, 28 in.; length of *main bore*, 448.5 in., diameter, 16 in.; *rifling*, 96 lands, 96 grooves; depth of groove, 0.06 in.; the

* Ordnance Department, U. S. A., "Notes on the Construction of Ordnance," No. 78.

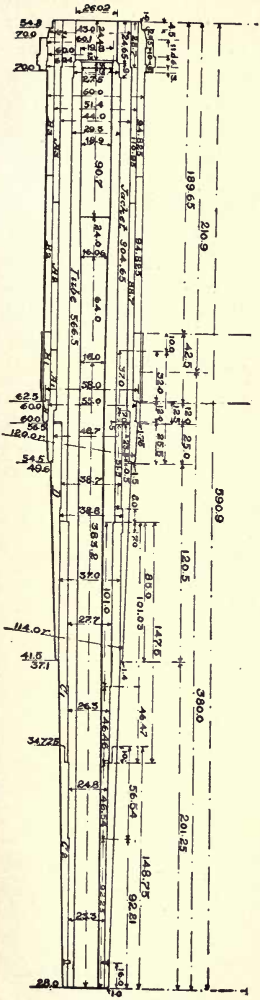


FIG. 19.

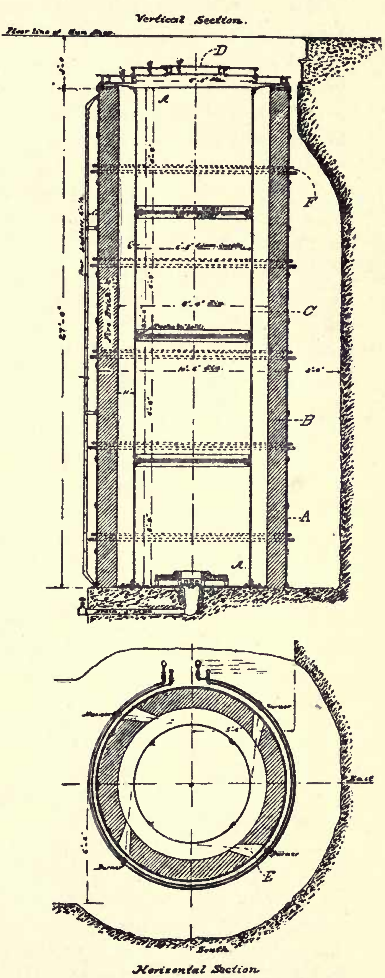


FIG. 20.

rifling curve is a semi-cubic parabola, ranging from one turn in 50 calibers to one in 25 at the muzzle. The cylindrical part of the *powder-chamber* is 90.7 in. long, and 18.9 in. diameter, and is connected with the bore by a conical slope 24 in. long. The volume of the chamber is 29,385 cu. in. The recess for breech-block is 24.4 in. long, with a diameter at top of thread of 24.86 in. The breech-mechanism is after the "Stockett System." The gun is built up of parts, as follows:

The *tube*, 566.5 in. long, with a maximum outside diameter of 29.3 in.; two *C-hoops* shrunk upon the tube from the forward end of the jacket to the muzzle; the *jacket*, 304.65 in. long, shrunk upon rear of tube, and overhanging the latter by 24.4 in. to form the breech-recess; the *D-hoop*, 144.5 in. long, encircling forward end of jacket and rear of *C*-hoop, and having two locking shoulders in its bore which engage corresponding projections on jacket and *C*-hoop, thus preventing any sliding backward of the former or forward of the latter, from the shock of firing; three *A-hoops*, *A-1* covering the joint between the *D*-hoop and the jacket, and *A-2*, *A-3*, being shrunk over the outer surface of the latter; four *B-hoops*, encircling the *A*-hoops.

Weights (lbs.).	Rough.	Finished.
Tube with *C*-hoops.	124,351	100,260
Jacket.	90,058	73,900
Hoop *D*.	26,965	23,900
" *A-1*.	19,859	14,910
" *A-2*.	16,137	15,120
" *A-3*.	20,163	19,940
" **B*,		
" **B-1*, *B-2*, *B-3*.	58,620	

The tube and jacket were each made from a nickel-steel ingot, not fluid-compressed, and octagon in section. After removing the discards, a longitudinal, axial hole was bored through the remaining block and the tube or jacket was then forged hollow on a mandrel under a hydraulic press. The completed forging was then rough-turned, bored, tempered in oil, and annealed. The hoops were made of fluid-compressed steel containing no nickel. Excepting that the ingots were round, the general process was similar to that for the tube and jacket. The hoop-metal was the harder, *i. e.*, having the greater elastic limit and tensile strength.

* Awaiting decision as to carriage.

All forgings were of sufficient total length to provide test-metal. The specimens for tube and jacket were 0.564 in. diameter and 3 in. long. The average physical qualities obtained in all tests are:

	Tube.	Jacket.	Hoops.
Elastic limit, lbs. per square inch.	51,375	52,250	57,125
Tensile strength, lbs. per square inch.	84,350	87,800	107,050
Elongation, per cent.	20.38	22.16	19.28
Contraction, " "	41.93	48.32	45.52

(*b*) *Shrinkage Furnace.*—The furnace used in expanding the parts for assemblage is shown in Fig. 20. It consists of a wrought-iron "cage" or frame-work *A*, surrounding immediately a cylindrical wall *B* of fire-brick, the whole resting upon solid rock, at the 30-ft. level, in a corner of the shrinkage-pit (Fig. 21). The thickness of the wall is 13 in. and its internal diameter is 8 ft. 4 in. A cylindrical muffle *C*, built of ½-in. boiler steel, surrounds the hoop to be heated. The outer diameter of the muffle is 6 ft. 6 in., there being, thus, an annular space, 11 in. wide, which forms a combustion-chamber for the burning gases. The furnace is 27 ft. 9 in. high; its top is 2 ft. 3 in. below the floor-level; it is closed by a removable cover *D*, which confines the steam and gases; and the products of combustion are drawn off through a flue connecting the top of the chamber with the main chimney.

Fuel oil is supplied through a 3-in. pipe from a 5,000-gallon tank and enters the furnaces through 20 burner-openings *E*, set in five tiers *F*, of four burners each. The burner consists of an internal steam-pipe of ¼-in. bore, the latter being reduced at the end to $\frac{1}{16}$ in. Surrounding this is a ½-in. oil-pipe, the forward end of which is plugged and a $\frac{1}{10}$-in. hole drilled therein, opposite the $\frac{1}{16}$-in. opening in the internal pipe. The steam issuing at high velocity through the latter opening, carries the oil with it as a spray; and its oxygen, combining with the oil, gives an intensely hot flame. The burners are so directed that the flame strikes the muffle at a tangent approximately, thus giving a rapid spiral movement to the gases. The muffle transmits the heat to the hoop and the circulation of air within it tends to make the temperature equal at all points of the hoop. The furnace-temperature is governed by a damper in the flue, by the number of jets burning, and by the amounts of oil and steam admitted. Each burner is surmounted by an observation opening, closed by a mica door.

Uniformity of heating is secured by the tangential direction of the gases and by the intervention of the muffle, the latter keeping the flames from impinging directly upon the hoop and thus causing local heating in excess.

(*c*) *Shrinkage-Pit.*—Within the same excavation which contains the shrinkage-furnace, the shrinkage-pit (Fig. 21) is located, the latter being 60 ft. deep and cut from the solid rock. To hold the gun during the shrinkage processes, a cast-iron chuck *G* is anchored in the concrete foundation at the bottom of the pit and an intermediate chuck *H* is placed at the 35-ft. level. Upon this level, also, there is constructed a heavy platform or "tipping rest" *K*, for supporting the lower end of the gun while it is lying in an angular position, after having been brought to, and partly lowered within, the pit by two cranes. The platform enables one of the latter to lift the gun to a vertical position and set it in the chucks. In order to handle the gun, when thus within the pit, two steel plugs, connected by a rod 7 in. in diameter and screwed

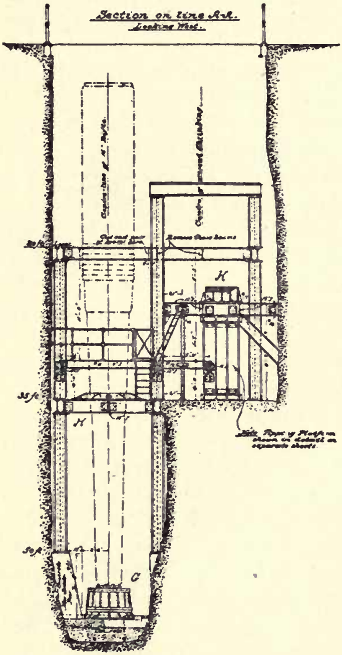


FIG. 21.

into each, were fitted within the bore of the tube, the plug at the upper end being arranged for connection with the bail on the crane-hook. A steam-pump to free the pit from the water used to cool hoops after assembling completes the equipment.

(*d*) *Assembling.*—In preparation for the shrinkage of the jacket, the tube was placed in the pit, muzzle-end down, and water connections were made for interior cooling and for cooling the jacket

when seated. The latter was then heated for 30 hours and its bore calipered three times during that period to determine the expansion. Upon removal from the furnace, it was measured, centered, and lowered in place and water was applied at the muzzle-end. The cooling continued for nine hours, the number of the encircling "water-rings" or pipes varying from four, as a maximum, to two at the close of the operation. The shrinkage of the *C*- and *D*-hoops was effected in a similar manner. The *A*-hoops were assembled with the gun in a horizontal position in the lathe. The hoists of a crane were attached to straps secured to the hoop after heating and the latter was carried to the gun, seated in place, and cooled by water from the forward end. During contraction, the hoop was under the constant pressure of two 30-ton hydraulic jacks, one on each side, acting in the horizontal plane through the axis of the gun. It is proposed to effect the seating of the *B*-hoops in a similar manner.

(*e*) *Expansion, Shrinkage, and Clearance.*—The expansion of the metal, per inch of diameter for each degree of temperature, was 0.000007 in. Thus, for an exterior diameter of hoop of 64 in., the total expansion for 1° of temperature = 0.000448 in., and, for 800°, = 0.358 in. Measured exterior diameters at several points on the surface of a hoop, if uniformly increased by expansion, indicate uniform temperature and the amount of expansion shows the degree of temperature. Calling the diameter of the cold tube D, that of the cold hoop or jacket d, and the shrinkage s:

$$\textit{Expansion} = 0.000007\,(D - s) = E;$$
$$\textit{Shrinkage} = D - d = s;$$
$$\textit{Clearance} = E + (D - s) - D;$$
$$\textit{Diameter of jacket heated} = E + (D - s).$$

CHAPTER II.

SCREW FASTENINGS.

A *screw-surface* or helicoid is described by a right line, *A-B*, Fig. 22, revolving about and advancing along an axis, *Y-Y*, as directrix, one extremity, *A*, of the line remaining upon the axis and the *angle of advance*, α, between the latter and the line or generatrix being constant. The *base-angle*, β, is the complement

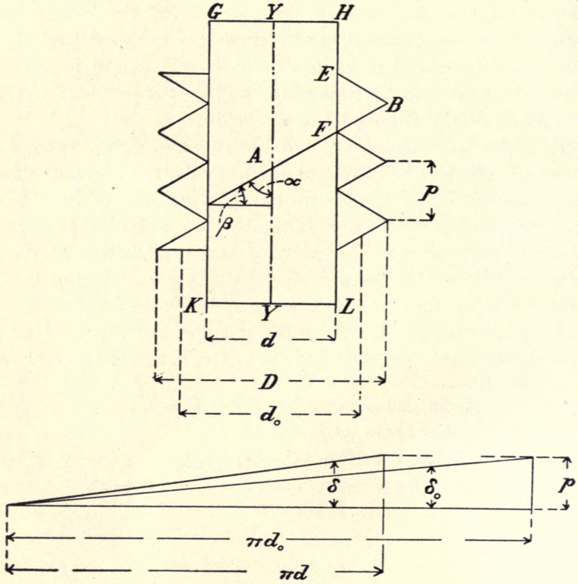


FIG. 22.

of the angle of advance. In the *screw-thread*, the generating line is replaced by a plane figure — as the triangle, *B-E-F*, a rectangle, or a trapezoid — maintained always in an axial plane and in contact with, and traversing a helical path upon, the surface of a cylinder, as *G-H-K-L*.

The *nominal*, or outer *diameter*, D, of a screw is that of the outside or top of the thread. The *effective diameter*, d, is that of the base or root of the thread and of the cylinder or core upon which the latter is described. The *depth of the thread* is the radial distance between its base and top, *i. e.*, $\frac{D-d}{2}$. The *pitch*, p, is the axial distance between adjacent convolutions of the *same* thread, *i. e.*, the axial distance which the nut traverses during one revolution. The *pitch-angle*, δ, of any helix of the thread, is the inclination between that helix and a plane perpendicular to the axis of the cylinder. While, in a normal screw, the pitch of all helices is the same, the pitch-angle of each depends upon its diameter. Calculations with regard to stresses within the thread are referred to the *mean thread-diameter*, d_0, (of pitch-angle δ_0), at which all forces are assumed to be concentrated. This diameter may be taken also, with sufficient accuracy, as that of the *mean helix*, equally distant from the helices at base and top of thread. The *projected area* of the thread is used in computations for bearing pressure.

In addition to differences in the forms of the threads, screws are distinguished further as *right- or left-handed* and *single- or multiple-threaded*. In a right-handed screw, the thread ascends contra-clockwise from left to right. Screw-fastenings have usually a single, right-handed, approximately triangular thread. A multiple (double, triple) threaded screw is one in which the cylinder is traversed by two or more threads, parallel and similar in all respects. Such screws, having ample bearing surface, are used for the transmission of power.

The screw and its nut form, kinematically, a pair; the relative motion of whose two elements consists of rotation about an axis and translation along the latter. If the material of the nut be relatively inelastic, as metal, the requirement for motion as above, is that the ratio between translation and rotation shall be constant, *i. e.*, that there shall be uniform pitch. When, however, the screw revolves in a mobile medium or nut, as water, its surface may have a varying pitch throughout. The marine propeller is a transverse section of a multiple-threaded screw, the pitch of whose blade-surface may be either constant or expand in either or both of two ways — radially outward or from the leading to the following edge of the blade.

11. Triangular vs. Square Threads.

The form of the thread is determined by the character of its service. The more important differences between the square thread and the full or modified triangular type lie in the relative strengths of these forms and the friction of operation. The load in a bolt is usually axial. It is transmitted to the bolt-thread and supported by the reaction of the nut-thread. The load-action and the nut-reaction must be, for equilibrium, equal. These mutual actions are, disregarding friction, normal to the contact surfaces, *i. e.*, to the threads. Considering friction, the reactions are diverted from the normal by the angle of friction, φ.

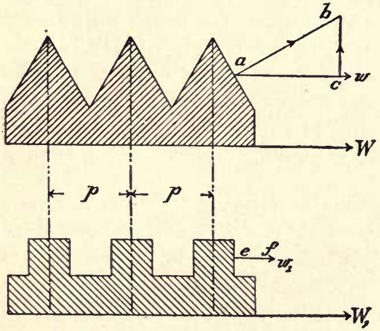


Fig. 23.

Fig. 23 represents sections of triangular and square-threaded bolts of the same pitch. Let W and W_1 be the axial loads respectively, n the number of threads in each nut, and $w = \frac{W}{n}$ and $w_1 = \frac{W_1}{n}$, the respective loads per thread. Disregarding the small angle, φ, the lines of action, *a-b* and *e-f* of the pressures due to the loads will be normal to the respective thread-surfaces. Consider the threads with regard to:

1. Friction. — This is directly proportional to the normal pressure upon the contact-surfaces. With the square thread, the unit-pressure upon the nut $= e\text{-}f$ and $\Sigma e\text{-}f = w_1$; but, with the triangular form, this unit-pressure $= a\text{-}b$, whose components are *a-c* and *b-c*. The latter acts to burst the nut while $\Sigma a\text{-}c = w$. Since $a\text{-}b > a\text{-}c$, there is, other things equal, greater friction with the triangular thread.

2. Strength. — In the triangular thread, the section at the root is the full length of the nut, while, in the square form, the section is but one half this length. Against shearing and flexure at the root, the latter thread is, therefore, proportionately the weaker.

3. Nut. — As noted, the triangular type has a bursting action upon the nut, which action, disregarding friction, does not exist with the square thread.

In general, the triangular form is more suitable for screw-fastenings, owing to its greater strength, its increased frictional holding power which prevents backing off under load, and the finer pitches permissible by the full section at the base of the thread. On the other hand, the square thread is better adapted for power-transmission, since it has less friction and its bursting effect upon the nut is so small as to be negligible.

12. Requirements of the Screw-Thread.

The screw is used as a *detachable fastening* in joining the members of a structure or machine; in *producing pressure or tension*, as in the screw-jack and testing-machine; and for the *transmission of power and conversion of motion*, as in the worm-gear and screw propeller. Its requirements for these uses are:

1. Power. — This depends upon the pitch and form. The effect of the latter upon the strength and power of thread has been discussed. With a given applied force, the less the pitch, the greater the axial load may be, since the pitch fixes the angle of the inclined plane upon which the load virtually moves.

2. Strength. — This is governed by the pitch, form and depth of the thread. With constant load, the steeper the pitch, the greater must be the applied power and the consequent normal pressure upon the thread. For the same load and nominal diameter, the deeper the thread, the less its mean bearing-pressure will be; but the moment of the load upon the root will be larger and the effective diameter of the bolt to resist tension, will be reduced.

3. Durability. — The most durable thread is one whose form produces the least friction, whose depth gives minimum bearing pressure, and which is most accurately fitted.

13. Elements of the Screw-Thread.

The requirements of the screw-thread make its elements interdependent. Consider:

1. Effective Diameter. — This depends upon the axial load and the torsional stress produced by friction between the threads in setting up the nut. The magnitude of the latter stress is gov-

erned by the applied power, and that of the power by the axial load and pitch.

2. PITCH. — The relations between pitch and diameter in the prevailing systems of screw-threads are the outcome less of logical analysis than of long experience. For screw-fastenings, the limit in one direction lies in the fact that, with an excessively coarse pitch, the depth will be too great and the effective diameter will be reduced unduly. Again, that component of the pressure which is parallel to the thread-surfaces will exceed the force of friction between the latter, and, owing to this excess, the nut will back off. On the other hand, with an unduly small pitch-angle, the surface-friction will form too large a proportion of the total work of setting up the nut, the torsional action upon the bolt will be excessive, and the latter may be sheared. In general, fine pitches are unsuitable for soft metals and coarse pitches for shallow holes.

3. FORM. — As stated, the square thread is the form best adapted for power-transmission. For large fastenings requiring to be readily and frequently removed and which are strained heavily, but in one direction only, as the breech-block of a gun, the trapezoidal thread (Fig. 30) is most suitable. This thread has the acting face normal to the axis, the rear face at an angle thereto, and combines the greatest strength and least friction attainable.

For screw-fastenings in general, the triangular thread, with blunt top, straight sides, and filled-in base-angle, was adopted through various considerations with regard to strength, friction, durability, ease of manufacture, and conformity with general practice. Thus, in strength and frictional holding power, this form is superior; its straight sides give even wear and maximum bearing surface; the angle between them is fixed, in the various systems, by compromises between the conditions as to strength, friction, bursting action upon the nut, and facility of verification and production; the flat or rounded top reduces the liability to injury; and the filling in of the reëntrant base angle increases the effective diameter of the bolt and, in the Seller's system, the resilience of the latter also.

4. NUT. — The nut may yield either by the shearing or rupture of its threads or by bursting from the action of the outward component of the pressure upon the thread. The latter, both on bolt and nut, acts as a cantilever beam, fixed at the root and loaded

uniformly over the bearing surface. When worn, the area of the latter is reduced, the bearing becomes irregular, the load is practically concentrated, and the bending moment at the root may be increased. If the nut is of a metal materially weaker than that of the bolt, its depth should be greater than the normal. In any event, this depth should be sufficient to give ample strength against flexure and shear at the root of the thread, to provide sufficient bearing surface to prevent abrasion, and to afford a good hold for the wrench.

5. Multiple Threads. — In power-transmission screws of large pitch, a single thread will provide adequate bearing surface only by having a depth so great as to give an unduly small effective diameter of bolt. When the pitch is sufficient to permit it, the use of two or more parallel threads of usual proportions will secure the required surface with a normal effective diameter. Such threads are usually of square or trapezoidal form.

14. The United States Standard (Sellers) Thread.

It would be difficult to overestimate the services to English-speaking engineers of Mr. William Sellers and of his predecessor in the same field, the late Sir Joseph Whitworth, in the investigations and efforts which led to the wide adoption of the respective systems of screw threads which bear their names. The two systems are in essentials almost identical. That of Sellers was originally presented by him before, and recommended by, the Franklin Institute in 1864. It was adopted later, with trifling modification, by the U. S. Navy and War Departments and by the Master Mechanics' and Master Car Builders' Associations and is now known as the U. S. Standard System of Screw Threads.

The thread, as shown in Fig. 24, is triangular with flat sides inclined at an angle of 60°, the apex being cut off and the base filled in to a radial distance in each case of one eighth the height of the primitive triangle making "flats," f, at these points each one eighth of the pitch, p, in length. The Sellers system provides dimensions for bolts from one fourth inch to six inches nominal diameter. The notation and formulæ are:

D = nominal (outside) diameter of bolt, inches;

d = effective diam., ins. $= D - 2s = D - 1.3p = D - \frac{1.3}{n}$;

$s =$ depth of thread, ins. $= \dfrac{D-d}{2} = p \times 0.65$; (31)

$p =$ pitch of thread, ins. $= 0.24\sqrt{D+0.625} - 0.175$; (32)

$n =$ number of threads per inch $= \dfrac{1}{p}$;

$f =$ width of flat $= \dfrac{p}{8}$; (33)

$H =$ depth of nut, rough $= D$;

$h =$ depth of head, rough $= \frac{1}{2} d_h$;

$d_n =$ short diam., hex. or square nut, rough $= \frac{3}{2} D + \frac{1}{8}''$;

$d_h =$ short diam. of head, rough $= \frac{3}{2} D + \frac{1}{8}''$;

The equation for the pitch, as above, is an empirical formula constructed to cover diameters within the scope of the system. To avoid impracticable fractions, the number of threads, as thus deduced, is modified to secure a convenient aliquot value. Thus, for a 2-in. bolt:

$$p = 0.24\sqrt{2+0.625} - 0.175 = 0.2138 \text{ in.};$$

$$1/0.2138 = 4.68 = n = \text{say, } 4.5 \text{ threads per in.}$$

The depth of the thread is obtained from the equation:

$$s = \tfrac{3}{4} p \cos 30^\circ = 0.65p,$$

deduced from the diagram, Fig. 24. The formula for the short diameter, d_n, of the nut is empirical and was derived from successful practice. The values of the depths, H and h, of the nut and head respectively were based upon considerations as to adequate bearing surface, shearing stress, and provision for an efficient hold for the wrench. The long diameters of hexagon and square figures may be obtained by multiplying the corresponding short diameters by 1.155 and 1.414, respectively. The finished dimensions for the depths and short diameters are:

H, finished $= D - \frac{1}{16}''$;

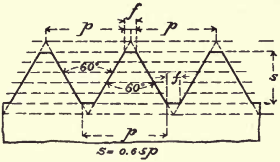
h, " $= D - \frac{1}{16}''$;

d_n, " $= \frac{3}{2} D + \frac{1}{16}''$;

d_h, " $= \frac{3}{2} D + \frac{1}{16}''$.

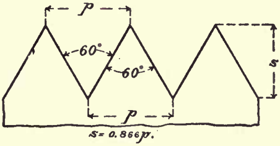
The U. S. Navy Department adopted the Sellers system with the single exception that no difference was made in the size

Fig. 24.



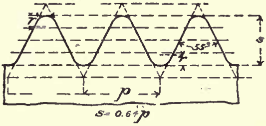
Sellers.

Fig. 25.



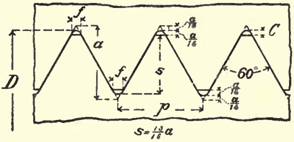
Sharp V.

Fig. 26.



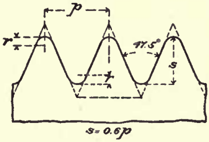
Whitworth.

Fig. 27.



Internat'l.

Fig. 28.



British Asso'n.

of finished and unfinished bolt-heads and nuts, in order that the same wrench might be used for both. The size adopted was that given by Sellers for rough work.

The formula for "the exact diameter of the tap-drill with no allowance for clearance is :

$$d = D - \frac{1.2990381}{n}.$$

"The usual allowance (for clearance) above exact bottom diameter is from 0.004 for $\frac{1}{4}$ inch to 0.010 for 2-inch taps." *

Table X.

U. S. Standard (Sellers) Bolts and Nuts.

Bolt.						Nut.	Head.	Nut and Head.
Diameter.		Area.		Threads.		Depth.	Depth.	
Nominal, D.	Effective, d.	Nominal, πD^2.	Effective, πd^2.	Number per Inch, n.	Width of flat, f.	H (rough).	h (rough).	Short Diameters dn, dh (rough).
$\frac{1}{4}$	0.185	0.049	0.027	20	0.0063	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{2}$
$\frac{5}{16}$	0.240	0.077	0.045	18	0.0069	$\frac{5}{16}$	$\frac{19}{64}$	$\frac{19}{32}$
$\frac{3}{8}$	0.294	0.110	0.068	16	0.0078	$\frac{3}{8}$	$\frac{11}{32}$	$\frac{11}{16}$
$\frac{7}{16}$	0.345	0.150	0.093	14	0.0089	$\frac{7}{16}$	$\frac{25}{64}$	$\frac{25}{32}$
$\frac{1}{2}$	0.400	0.196	0.126	13	0.0096	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{8}$
$\frac{9}{16}$	0.454	0.249	0.162	12	0.0104	$\frac{9}{16}$	$\frac{31}{64}$	$\frac{31}{32}$
$\frac{5}{8}$	0.507	0.307	0.202	11	0.0114	$\frac{5}{8}$	$\frac{17}{32}$	$1\frac{1}{16}$
$\frac{3}{4}$	0.620	0.442	0.302	10	0.0125	$\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{4}$
$\frac{7}{8}$	0.731	0.601	0.420	9	0.0139	$\frac{7}{8}$	$\frac{23}{32}$	$1\frac{7}{16}$
1	0.838	0.785	0.550	8	0.0156	1	$\frac{13}{16}$	$1\frac{5}{8}$
$1\frac{1}{8}$	0.939	0.994	0.694	7	0.0179	$1\frac{1}{8}$	$\frac{29}{32}$	$1\frac{13}{16}$
$1\frac{1}{4}$	1.064	1.227	0.893	7	0.0179	$1\frac{1}{4}$	1	2
$1\frac{3}{8}$	1.159	1.485	1.057	6	0.0208	$1\frac{3}{8}$	$1\frac{3}{32}$	$2\frac{3}{16}$
$1\frac{1}{2}$	1.284	1.767	1.295	6	0.0208	$1\frac{1}{2}$	$1\frac{3}{16}$	$2\frac{3}{8}$
$1\frac{5}{8}$	1.389	2.074	1.515	$5\frac{1}{2}$	0.0227	$1\frac{5}{8}$	$1\frac{9}{32}$	$2\frac{9}{16}$
$1\frac{3}{4}$	1.490	2.405	1.746	5	0.0250	$1\frac{3}{4}$	$1\frac{3}{8}$	$2\frac{3}{4}$
$1\frac{7}{8}$	1.615	2.761	2.051	5	0.0250	$1\frac{7}{8}$	$1\frac{15}{32}$	$2\frac{15}{16}$
2	1.711	3.142	2.302	$4\frac{1}{2}$	0.0278	2	$1\frac{9}{16}$	$3\frac{1}{8}$
$2\frac{1}{4}$	1.961	3.976	3.023	$4\frac{1}{2}$	0.0278	$2\frac{1}{4}$	$1\frac{3}{4}$	$3\frac{1}{2}$
$2\frac{1}{2}$	2.175	4.909	3.719	4	0.0313	$2\frac{1}{2}$	$1\frac{15}{16}$	$3\frac{7}{8}$
$2\frac{3}{4}$	2.425	5.940	4.620	4	0.0313	$2\frac{3}{4}$	$2\frac{1}{8}$	$4\frac{1}{4}$
3	2.629	7.069	5.428	$3\frac{1}{2}$	0.0357	3	$2\frac{5}{16}$	$4\frac{5}{8}$
$3\frac{1}{4}$	2.879	8.296	6.510	$3\frac{1}{2}$	0.0357	$3\frac{1}{4}$	$2\frac{1}{2}$	5
$3\frac{1}{2}$	3.100	9.621	7.548	$3\frac{1}{4}$	0.0385	$3\frac{1}{2}$	$2\frac{11}{16}$	$5\frac{3}{8}$
$3\frac{3}{4}$	3.317	11.045	8.641	3	0.0417	$3\frac{3}{4}$	$2\frac{7}{8}$	$5\frac{3}{4}$
4	3.567	12.566	9.963	3	0.0417	4	$3\frac{1}{16}$	$6\frac{1}{8}$
$4\frac{1}{4}$	3.798	14.186	11.329	$2\frac{7}{8}$	0.0435	$4\frac{1}{4}$	$3\frac{1}{4}$	$6\frac{1}{2}$
$4\frac{1}{2}$	4.028	15.904	12.753	$2\frac{3}{4}$	0.0455	$4\frac{1}{2}$	$3\frac{7}{16}$	$6\frac{7}{8}$
$4\frac{3}{4}$	4.255	17.721	14.226	$2\frac{5}{8}$	0.0476	$4\frac{3}{4}$	$3\frac{5}{8}$	$7\frac{1}{4}$
5	4.480	19.635	15.763	$2\frac{1}{2}$	0.0500	5	$3\frac{13}{16}$	$7\frac{5}{8}$
$5\frac{1}{4}$	4.730	21.648	17.572	$2\frac{1}{2}$	0.0500	$5\frac{1}{4}$	4	8
$5\frac{1}{2}$	4.953	23.758	19.267	$2\frac{3}{8}$	0.0526	$5\frac{1}{2}$	$4\frac{3}{16}$	$8\frac{3}{8}$
$5\frac{3}{4}$	5.203	25.967	21.262	$2\frac{3}{8}$	0.0526	$5\frac{3}{4}$	$4\frac{3}{8}$	$8\frac{3}{4}$
6	5.423	28.274	23.098	$2\frac{1}{4}$	0.0556	6	$4\frac{9}{16}$	$9\frac{1}{8}$

* "Standards of Length," G. M. Bond, 1887, p. 169.

The Sellers system was investigated exhaustively by a Board of U. S. Naval Engineer officers in 1868. This Board * found as to

1. *Pitch.* The relations of pitch and diameter did not differ materially from the average proportions dictated by good practice. 2. *Form.* The thread, as compared with that of ordinary V form, gave with equal pitches a greater effective diameter and was less liable to injury. Further, in the most unfavorable case — that of the one-fourth-inch bolt — where the inclination of the thread and the torsional stress are maxima, the tendencies of the bolt to yield to tension or torsion are, with clean and well-lubricated surfaces, about equal. 3. *Nut.* To resist shearing (stripping) of the thread, the depth, $H = D$, gives a marked excess of strength for perfect threads, since, for the latter, but $0.357d$ is required. With regard to bearing surface for fastenings, the depth, H, provides as much or more than nuts were given ordinarily. The diameter, d_n, was found to give ample security against bursting action, since, neglecting the resistance of the thread and taking the entire section of the bolt as effective, the required diameter, $d_n = 1\frac{1}{6}D$. 4. *Head.* The depth, h, was sufficient to provide fully against shearing and to afford an efficient hold for the wrench.

The proportions of the Sellers system are given in Table X.

15. Modifications of the Sellers System.

Experience with the proportions of this system has resulted in modifications as to:

1. Pitch and Diameter. — For nominal diameters ranging from 2¾ ins. to 6 ins., equation (32) gives the corresponding numbers of threads per inch as ranging from 4 at 2¾ in. to 2¼ at 6 in. These proportions, theoretically, should be such as will give a bolt equally strong in all respects. In naval practice and in that of many large companies, it is now customary to make the number of threads per inch 4 for all diameters from 2½ in. to 6 in., inclusive, thus increasing materially the effective diameter of the bolt. The proportions of bolts and nuts now prescribed by the Bureau of Steam Engineering, U. S. Navy, are given in Table XI.

2. Bolt-Heads and Nuts. — The proportions of nuts and bolt-heads, as given in the Sellers system, require odd sizes of bar-metal, not usually rolled by the mills, for the nuts and addi-

* "Report of Board to Recommend a Standard Gauge for Bolts, Nuts, and Screw-Threads for U. S. Navy." May, 1868.

tional upsets in order to obtain sufficient metal for the standard head. Tables XII. and XIII. give dimensions which are without these disadvantages.

3. CIRCULAR NUTS. — The Sellers system gives the dimensions of hexagonal and square nuts only. The former are lighter, their long diameter is less, and, where the movement of the wrench is restricted, they are more readily screwed home. The circular, grooved nut is a form applicable for use in a confined space and is of especial value where very large sizes are required as, for example, on the end of a propeller shaft. The outside diameter of the circular nut is equal to the short diameter of the other types,

TABLE XI.

STANDARD DIMENSIONS OF BOLTS AND NUTS FOR U. S. NAVY.
(BUREAU OF STEAM ENGINEERING.)

Diam. D.	Eff. Diam. $D-d$.	Threads Per Inch.	Long Diam.		Short Diam.	Depth.	
			Hex.	Sq.		Head.	Nut.
$\frac{1}{4}''$	.065″	20	$\frac{9}{16}''$	$\frac{23}{32}''$	$\frac{1}{2}''$	$\frac{1}{4}''$	$\frac{1}{4}''$
$\frac{5}{16}$	.072	18	$\frac{11}{16}$	$\frac{27}{32}$	$\frac{19}{32}$	$\frac{19}{64}$	$\frac{5}{16}$
$\frac{3}{8}$	.081	16	$\frac{25}{32}$	$\frac{31}{32}$	$\frac{11}{16}$	$\frac{11}{32}$	$\frac{3}{8}$
$\frac{7}{16}$	.093	14	$\frac{29}{32}$	$1\frac{3}{32}$	$\frac{25}{32}$	$\frac{25}{64}$	$\frac{7}{16}$
$\frac{1}{2}$	.100	13	1	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
$\frac{9}{16}$	.108	12	$1\frac{1}{8}$	$1\frac{3}{8}$	$\frac{31}{32}$	$\frac{31}{64}$	$\frac{9}{16}$
$\frac{5}{8}$	.118	11	$1\frac{7}{32}$	$1\frac{1}{2}$	$1\frac{1}{16}$	$\frac{17}{32}$	$\frac{5}{8}$
$\frac{3}{4}$	.130	10	$1\frac{7}{16}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{7}{8}$	.144	9	$1\frac{21}{32}$	$2\frac{1}{32}$	$1\frac{7}{16}$	$\frac{23}{32}$	$\frac{7}{8}$
1	.162	8	$1\frac{7}{8}$	$2\frac{5}{16}$	$1\frac{5}{8}$	$\frac{13}{16}$	1
$1\frac{1}{8}$	.186	7	$2\frac{3}{32}$	$2\frac{9}{16}$	$1\frac{13}{16}$	$\frac{29}{32}$	$1\frac{1}{8}$
$1\frac{1}{4}$	.186	7	$2\frac{5}{16}$	$2\frac{27}{32}$	2	1	$1\frac{1}{4}$
$1\frac{3}{8}$	.217	6	$2\frac{17}{32}$	$3\frac{3}{32}$	$2\frac{3}{16}$	$1\frac{3}{32}$	$1\frac{3}{8}$
$1\frac{1}{2}$	.217	6	$2\frac{3}{4}$	$3\frac{11}{32}$	$2\frac{3}{8}$	$1\frac{3}{16}$	$1\frac{1}{2}$
$1\frac{5}{8}$	.236	$5\frac{1}{2}$	$2\frac{31}{32}$	$3\frac{5}{8}$	$2\frac{9}{16}$	$1\frac{9}{32}$	$1\frac{5}{8}$
$1\frac{3}{4}$	.260	5	$3\frac{3}{16}$	$3\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{3}{8}$	$1\frac{3}{4}$
$1\frac{7}{8}$	.260	5	$3\frac{13}{32}$	$4\frac{5}{32}$	$2\frac{15}{16}$	$1\frac{15}{32}$	$1\frac{7}{8}$
2	.289	$4\frac{1}{2}$	$3\frac{19}{32}$	$4\frac{13}{32}$	$3\frac{1}{8}$	$1\frac{9}{16}$	2
$2\frac{1}{4}$	.289	$4\frac{1}{2}$	$4\frac{1}{32}$	$4\frac{15}{16}$	$3\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$
$2\frac{1}{2}$	.325	4	$4\frac{15}{32}$	$5\frac{15}{32}$	$3\frac{7}{8}$	$1\frac{15}{16}$	$2\frac{1}{2}$
$2\frac{3}{4}$	"	4	$4\frac{29}{32}$	6	$4\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{3}{4}$
3	"	4	$5\frac{11}{32}$	$6\frac{17}{32}$	$4\frac{5}{8}$	$2\frac{5}{16}$	3
$3\frac{1}{4}$	"	4	$5\frac{25}{32}$	$7\frac{1}{16}$	5	$2\frac{1}{2}$	$3\frac{1}{4}$
$3\frac{1}{2}$	"	4	$6\frac{7}{32}$	$7\frac{19}{32}$	$5\frac{3}{8}$	$2\frac{11}{16}$	$3\frac{1}{2}$
$3\frac{3}{4}$	"	4	$6\frac{5}{8}$	$8\frac{1}{8}$	$5\frac{3}{4}$	$2\frac{7}{8}$	$3\frac{3}{4}$
4	"	4	$7\frac{1}{16}$	$8\frac{21}{32}$	$6\frac{1}{8}$	$3\frac{1}{16}$	4
$4\frac{1}{4}$	"	4	$7\frac{1}{2}$	$9\frac{3}{16}$	$6\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{4}$
$4\frac{1}{2}$	"	4	$7\frac{15}{16}$	$9\frac{23}{32}$	$6\frac{7}{8}$	$3\frac{7}{16}$	$4\frac{1}{2}$
$4\frac{3}{4}$	"	4	$8\frac{3}{8}$	$10\frac{1}{4}$	$7\frac{1}{4}$	$3\frac{5}{8}$	$4\frac{3}{4}$
5	"	4	$8\frac{13}{16}$	$10\frac{25}{32}$	$7\frac{5}{8}$	$3\frac{13}{16}$	5
$5\frac{1}{4}$	"	4	$9\frac{1}{4}$	$11\frac{5}{16}$	8	4	$5\frac{1}{4}$
$5\frac{1}{2}$	"	4	$9\frac{11}{16}$	$11\frac{27}{32}$	$8\frac{3}{8}$	$4\frac{3}{16}$	$5\frac{1}{2}$
$5\frac{3}{4}$	"	4	$10\frac{3}{32}$	$12\frac{3}{8}$	$8\frac{3}{4}$	$4\frac{3}{8}$	$5\frac{3}{4}$
6	"	4	$10\frac{17}{32}$	$12\frac{29}{32}$	$9\frac{1}{8}$	$4\frac{9}{16}$	6

plus twice the depth of the groove. In large sizes, this diameter is less than the long diameter of the hexagonal form. Good proportions for circular nuts are given in Table XIV.

Table XII.

Manufacturers' Standard Dimensions of Bolt Heads.
(American Iron and Steel Manufacturing Company.)

Diameter, Bolt.	Square and Hexagon Heads, Width and Thickness.	Diameter, Bolt.	Square and Hexagon Heads, Width and Thickness.
1/4	3/8 × 3/16	1	1 1/2 × 7/8
5/16	15/32 × 1/4	1 1/8	1 11/16 × 1
3/8	9/16 × 5/16	1 1/4	1 7/8 × 1 1/8
7/16	21/32 × 3/8	1 3/8	2 1/16 × 1 1/4
1/2	3/4 × 7/16	1 1/2	2 1/4 × 1 3/8
9/16	27/32 × 1/2	1 5/8	2 7/16 × 1 1/2
5/8	15/16 × 17/32	1 3/4	2 5/8 × 1 5/8
3/4	1 1/8 × 5/8	1 7/8	2 13/16 × 1 3/4
7/8	1 5/16 × 3/4	2	3 × 1 7/8

Table XIII.

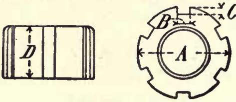
Manufacturers' Standard Dimensions of Hot-Pressed Nuts.
(American Iron and Steel Manufacturing Company.)

Square.				Hexagon.			
Short Dia.	Thickness.	Hole.	Size, Bolt.	Short Dia.	Thickness.	Hole.	Size, Bolt.
1/2	1/4	7/32	1/4	1/2	1/4	7/32	1/4
5/8	5/16	9/32	5/16	5/8	5/16	9/32	5/16
3/4	3/8	11/32	3/8	3/4	3/8	11/32	3/8
7/8	7/16	25/64	7/16	7/8	7/16	25/64	7/16
1	1/2	7/16	1/2	1	1/2	7/16	1/2
1 1/8	9/16	1/2	9/16	1 1/8	9/16	1/2	9/16
1 1/4	5/8	9/16	5/8	1 1/4	5/8	9/16	5/8
1 1/2	3/4	21/32	3/4	1 3/8	3/4	21/32	3/4
1 3/4	7/8	25/32	7/8	1 5/8	7/8	25/32	7/8
2	1	7/8	1	1 3/4	1	7/8	1
2 1/4	1 1/8	31/32	1 1/8	2	1 1/4	31/32	1 1/8
2 1/2	1 1/4	1 3/32	1 1/4	2 1/4	1 3/8	1 3/32	1 1/4
2 3/4	1 3/8	1 3/16	1 3/8	2 1/2	1 1/2	1 3/16	1 3/8
3	1 1/2	1 5/16	1 1/2	2 3/4	1 5/8	1 5/16	1 1/2
3 1/4	1 5/8	1 7/16	1 5/8	3	1 3/4	1 7/16	1 5/8
3 1/2	1 3/4	1 9/16	1 3/4	3 1/4	1 7/8	1 9/16	1 3/4
3 3/4	1 7/8	1 11/16	1 7/8	3 1/2	2	1 11/16	1 7/8
4	2	1 13/16	2	3 1/2	2	1 13/16	2
4	2 1/8	1 7/8	2 1/8	3 3/4	2 1/8	1 7/8	2 1/8
4 1/4	2 1/4	2	2 1/4	3 3/4	2 1/4	2	2 1/4
4 1/4	2 3/8	2 1/8	2 3/8	4	2 3/8	2 1/8	2 3/8
4 1/2	2 1/2	2 1/4	2 1/2	4 1/4	2 1/2	2 1/4	2 1/2
4 3/4	2 3/4	2 7/16	2 3/4	4 1/2	2 3/4	2 7/16	2 3/4
5	3	2 11/16	3	4 3/4	3	2 11/16	3
5 1/2	3 1/4	2 15/16	3 1/4	5	3 1/4	2 15/16	3 1/4
6	3 1/2	3 1/8	3 1/2	5 1/4	3 1/2	3 1/8	3 1/2

Table XIV.

Round Slotted Nuts.

(Newport News Shipbuilding and Dry Dock Company.)



Diameter of Bolt.	A	B	C	D	Diameter of Bolt.	A	B	C	D
$\frac{3}{4}''$	$1\frac{5}{8}''$	$\frac{5}{16}''$	$\frac{1}{8}''$	$\frac{3}{4}''$	$6''$	$10\frac{1}{4}''$	$1\frac{1}{4}''$	$\frac{1}{2}''$	$6''$
$\frac{7}{8}$	$1\frac{7}{8}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{7}{8}$	$6\frac{1}{4}$	$10\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$6\frac{1}{4}$
1	2	$\frac{3}{8}$	$\frac{3}{16}$	1	$6\frac{1}{2}$	11	$1\frac{3}{8}$	$\frac{9}{16}$	$6\frac{1}{2}$
$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	$1\frac{1}{8}$	$6\frac{3}{4}$	$11\frac{1}{2}$	$1\frac{3}{8}$	$\frac{9}{16}$	$6\frac{3}{4}$
$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$1\frac{1}{4}$	7	$11\frac{7}{8}$	$1\frac{3}{8}$	$\frac{9}{16}$	7
$1\frac{3}{8}$	$2\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{16}$	$1\frac{3}{8}$	$7\frac{1}{4}$	$12\frac{1}{4}$	$1\frac{1}{2}$	$\frac{9}{16}$	$7\frac{1}{4}$
$1\frac{1}{2}$	$2\frac{7}{8}$	$\frac{7}{16}$	$\frac{3}{16}$	$1\frac{1}{2}$	$7\frac{1}{2}$	$12\frac{5}{8}$	$1\frac{1}{2}$	$\frac{5}{8}$	$7\frac{1}{2}$
$1\frac{5}{8}$	$3\frac{1}{8}$	$\frac{7}{16}$	$\frac{3}{16}$	$1\frac{5}{8}$	$7\frac{3}{4}$	13	$1\frac{1}{2}$	$\frac{5}{8}$	$7\frac{3}{4}$
$1\frac{3}{4}$	$3\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{16}$	$1\frac{3}{4}$	8	$13\frac{1}{2}$	$1\frac{1}{2}$	$\frac{5}{8}$	8
$1\frac{7}{8}$	$3\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$	$1\frac{7}{8}$	$8\frac{1}{4}$	$13\frac{7}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$8\frac{1}{4}$
2	$3\frac{5}{8}$	$\frac{1}{2}$	$\frac{1}{4}$	2	$8\frac{1}{2}$	$14\frac{1}{4}$	$1\frac{5}{8}$	$\frac{11}{16}$	$8\frac{1}{2}$
$2\frac{1}{4}$	$4\frac{1}{8}$	$\frac{9}{16}$	$\frac{1}{4}$	$2\frac{1}{4}$	$8\frac{3}{4}$	$14\frac{3}{4}$	$1\frac{5}{8}$	$\frac{11}{16}$	$8\frac{3}{4}$
$2\frac{1}{2}$	$4\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$	9	$15\frac{1}{8}$	$1\frac{3}{4}$	$\frac{11}{16}$	9
$2\frac{3}{4}$	$4\frac{7}{8}$	$\frac{5}{8}$	$\frac{5}{16}$	$2\frac{3}{4}$	$9\frac{1}{4}$	$15\frac{1}{2}$	$1\frac{3}{4}$	$\frac{11}{16}$	$9\frac{1}{4}$
3	$5\frac{1}{4}$	$\frac{11}{16}$	$\frac{5}{16}$	3	$9\frac{1}{2}$	16	$1\frac{3}{4}$	$\frac{3}{4}$	$9\frac{1}{2}$
$3\frac{1}{4}$	$5\frac{5}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	$3\frac{1}{4}$	$9\frac{3}{4}$	$16\frac{3}{8}$	$1\frac{7}{8}$	$\frac{3}{4}$	$9\frac{3}{4}$
$3\frac{1}{2}$	$6\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	$3\frac{1}{2}$	10	$16\frac{3}{4}$	$1\frac{7}{8}$	$\frac{3}{4}$	10
$3\frac{3}{4}$	$6\frac{1}{2}$	$\frac{13}{16}$	$\frac{3}{8}$	$3\frac{3}{4}$	$10\frac{1}{4}$	$17\frac{1}{8}$	$1\frac{7}{8}$	$\frac{13}{16}$	$10\frac{1}{4}$
4	$6\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	4	$10\frac{1}{2}$	$17\frac{1}{2}$	$1\frac{7}{8}$	$\frac{13}{16}$	$10\frac{1}{2}$
$4\frac{1}{4}$	$7\frac{3}{8}$	$\frac{15}{16}$	$\frac{3}{8}$	$4\frac{1}{4}$	$10\frac{3}{4}$	18	2	$\frac{13}{16}$	$10\frac{3}{4}$
$4\frac{1}{2}$	$7\frac{3}{4}$	1	$\frac{3}{8}$	$4\frac{1}{2}$	11	$18\frac{3}{8}$	2	$\frac{7}{8}$	11
$4\frac{3}{4}$	$8\frac{1}{8}$	1	$\frac{7}{16}$	$4\frac{3}{4}$	$11\frac{1}{4}$	$18\frac{3}{4}$	$2\frac{1}{8}$	$\frac{7}{8}$	$11\frac{1}{4}$
5	$8\frac{1}{2}$	$1\frac{1}{8}$	$\frac{7}{16}$	5	$11\frac{1}{2}$	$19\frac{1}{4}$	$2\frac{1}{8}$	$\frac{7}{8}$	$11\frac{1}{2}$
$5\frac{1}{4}$	9	$1\frac{1}{8}$	$\frac{7}{16}$	$5\frac{1}{4}$	$11\frac{3}{4}$	$19\frac{5}{8}$	$2\frac{1}{4}$	$\frac{7}{8}$	$11\frac{3}{4}$
$5\frac{1}{2}$	$9\frac{3}{8}$	$1\frac{1}{8}$	$\frac{7}{16}$	$5\frac{1}{2}$	12	20	$2\frac{1}{4}$	$\frac{7}{8}$	12
$5\frac{3}{4}$	$9\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$5\frac{3}{4}$					

16. The Sharp "V" Thread.

This thread has been superseded very largely in the United States by that of Sellers. As shown in Fig. 25, the sides are inclined to each other at an angle of 60° and have a sharp apex and base. A section of the thread forms, therefore, an equilateral triangle, each side of which is equal to the pitch of the screw. Using previous notation :

$$s = p \cos 30° = 0.866p;$$

$$d = D - 2s = D - 1.732p;$$

$$p = \frac{1}{n}.$$

The pitch is usually that of the Sellers system.

17. The Whitworth System of Screw-Threads.

In 1841 the late Sir Joseph Whitworth brought forward, in a communication to the Institution of Civil Engineers, the system of screw-threads which bears his name. This system, modified slightly in 1857 and 1861, has met universal adoption in Great Britain and extended use upon the continent of Europe. The range of diameters was originally, as in the Sellers system, from one quarter inch to six inches.

TABLE XV.

WHITWORTH SYSTEM. BOLTS AND NUTS.

Bolt.					Hexagon.	
Diameter.		Area.	Pitch.	Threads.	Head.	Nut and Head.
Nominal, D.	Effective, d.	Effective, πd^2.	p.	Per Inch, n.	Depth, h.	Short Diam., d_n and d_h.
$\frac{1}{4}$	0.186	0.0272	0.0500	20	0.2187	0.525
$\frac{5}{16}$	0.241	0.0456	0.0555	18	0.2734	0.601
$\frac{3}{8}$	0.295	0.0683	0.0625	16	0.3281	0.709
$\frac{7}{16}$	0.346	0.0940	0.0714	14	0.3828	0.820
$\frac{1}{2}$	0.393	0.1213	0.0833	12	0.4375	0.920
$\frac{5}{8}$	0.508	0.2035	0.0909	11	0.5468	1.100
$\frac{3}{4}$	0.622	0.3038	0.1000	10	0.6562	1.300
$\frac{7}{8}$	0.732	0.4219	0.1110	9	0.7656	1.480
1	0.840	0.5542	0.1250	8	0.8750	1.670
$1\frac{1}{8}$	0.942	0.6969	0.1428	7	0.9843	1.860
$1\frac{1}{4}$	1.067	0.8942	0.1428	7	1.0937	2.050
$1\frac{3}{8}$	1.161	1.0597	0.1666	6	1.2031	2.210
$1\frac{1}{2}$	1.286	1.3009	0.1666	6	1.3125	2.410
$1\frac{5}{8}$	1.368	1.4719	0.2000	5	1.4128	2.570
$1\frac{3}{4}$	1.494	1.7530	0.2000	5	1.5312	2.750
$1\frac{7}{8}$	1.590	1.9855	0.2222	$4\frac{1}{2}$	1.6406	3.020
2	1.715	2.3101	0.2222	$4\frac{1}{2}$	1.7500	3.150
$2\frac{1}{4}$	1.930	2.9255	0.2500	4	1.9687	3.540
$2\frac{1}{2}$	2.180	3.7325	0.2500	4	2.1875	3.890
$2\frac{3}{4}$	2.384	4.4637	0.2857	$3\frac{1}{2}$	2.4062	4.180
3	2.634	5.4490	0.2857	$3\frac{1}{2}$	2.6250	4.530
$3\frac{1}{4}$	2.856	6.4063	0.3077	$3\frac{1}{4}$	2.8256	4.850
$3\frac{1}{2}$	3.105	7.5769	0.3077	$3\frac{1}{4}$	3.0624	5.170
$3\frac{3}{4}$	3.320	8.6726	0.3333	3	3.2812	5.550
4	3.573	10.0270	0.3333	3	3.5000	5.950
$4\frac{1}{4}$	3.804	11.3710	0.3478	$2\frac{7}{8}$	3.7046	6.370
$4\frac{1}{2}$	4.054	12.9140	0.3478	$2\frac{7}{8}$	3:9374	6.820
$4\frac{3}{4}$	4.284	14.4140	0.3636	$2\frac{3}{4}$	4.1562	7.300
5	4.534	16.1460	0.3636	$2\frac{3}{4}$	4.3750	7.800
$5\frac{1}{4}$	4.762	17.8100	0.3809	$2\frac{5}{8}$	4.5936	8.350
$5\frac{1}{2}$	5.012	19.7290	0.3809	$2\frac{5}{8}$	4.8124	8.850
$5\frac{3}{4}$	5.240	21.5490	0.4000	$2\frac{1}{2}$	5.0312	9.450
6	5.487	23.6540	0.4000	$2\frac{1}{2}$	5.2500	10.000

As shown in Fig. 26, the thread is triangular in section, the angle between the sides being 55°. The primitive triangle

is rounded off at the top and bottom by an amount equal, in each case, to one sixth of its height, making the depth of the thread two thirds of the altitude. The relation between diameter and pitch, the angle of the sides, and the depth of the thread were determined by taking the mean of the variations in these respects of a large collection of screw-bolts gathered from the principal machine-shops throughout England. The one quarter inch, one half inch, one inch, and one and one half inch bolts were examined particularly and taken as the fixed points of a scale by which intermediate sizes were regulated, deviation from the exact average being made only to avoid small fractional parts in the number of threads per inch. The formulæ are:

$$s = \tfrac{2}{3}p \div \tan 27^\circ\ 30' = 0.64\,p;$$

$$d = D - 2s = D - 1.28\,p;$$

$$p = \frac{1}{n} = 0.08\ D + 0.04 \text{ approximately};$$

$$r = \text{radius of rounding} = 0.1373\,p.$$

The dimensions of the system are given in Table XV. The depth of the nut is equal to the nominal diameter of the bolt.

18. The Sharp V, Sellers, and Whitworth Threads.

Consider bolts of the same nominal diameter in these systems with regard to:

1. TENSILE STRENGTH.—The effective diameters are:

V Thread, $d = D - 1.732p$;

Sellers, $d = D - 1.3p$;

Whitworth, $d = D - 1.28p$.

2. STRIPPING OF THREAD.—The section, at base of thread, to resist shear in:

V and Whitworth Threads $= p$;

Sellers Threads $= 0.875p$.

3. BEARING SURFACE.—This is a maximum in the V thread with its straight sides from apex to root and a minimum in the Whitworth form owing to the rounding. The Sellers thread holds an intermediate place.

4. FRICTION.—The normal pressure and, therefore, the friction are less in the Whitworth thread than in the other types, owing to the smaller angle of the sides.

5. RESILIENCE.—The section of least diameter is but a line in the V thread and is a flat, $\frac{1}{8}p$ in length, in the Sellers system. The rounded base gives the Whitworth form an intermediate position. While the Sellers type seems thus to be superior, the sudden change in section at the bottom of its thread is a source of weakness.

6. DURABILITY.—The sharp tops of the V thread are very liable to injury. In this and the Sellers form, the normal pressure is uniform over the entire surface, while, in the Whitworth thread, it is uniform upon the sides and varying and greater over the curved surfaces. The wear of the Sellers type will be, therefore, less than that of the Whitworth.

7. REPRODUCTION.—The 60° angle can be reproduced and verified more readily than one of 55°. The curves in the Whitworth form vary in radius with the pitch and cannot be made with the same degree of precision as the flats of the Sellers system. The taps and dies used in the making of the V thread soon lose their fine cutting edges, thus causing constant variations in fitting.

19. French Standard Screw-Thread.

(*Système Unifié Français.*)

To the *Société d'Encouragement pour l'Industrie Nationale* is due the credit for the adoption of a standard thread in France. The thread form is practically that of Sellers based on metric units. The section is an equilateral triangle whose base is equal to the pitch, the top of the triangle being cut off and the root of the thread filled in to form flats, situated one eighth the height of the triangle from its apex and base respectively. As in the Sellers system:

$$\text{Angle} = 60° ;$$

$$\text{Depth } s = 0.65\, p ;$$

$$\text{Width of flat, } f = \frac{p}{8}.$$

The proportions of the system are given, in millimetres, in Table XVI. It has been extended to a nominal diameter of 148 mm. (5.82 in.) and a pitch of 10.5 mm. (0.4133 in.). At nominal

diameters of 80, 96, 106, 116, 126, 136 and 148 mm., the pitches, respectively, are: 7, 8, 8.5, 9, 9.5, 10, 10.5 mm. The standard screws adopted by the French Navy include the extended series as above with certain others intercalated to meet the requirements of the service.

TABLE XVI.

FRENCH STANDARD SCREW-THREADS.

Diameter.		Thread.	Pitch.
Nominal,* D.	Effective, d.	Depth, s.	p.
mm.	mm.	mm.	mm.
6	4.70	0.650	1.0
10	8.50	0.975	1.5
14	11.40	1.300	2.0
18	14.75	1.625	2.5
24	20.10	1.950	3.0
30	25.45	2.275	3.5
36	30.80	2.600	4.0
42	36.15	2.925	4.5
48	41.50	3.250	5.0
56	48.85	3.575	5.5
64	56.20	3.900	6.0
72	63.55	4.225	6.5
80	70.90	4.550	7.0

20. International Standard Screw-Thread.

(*Système Internationale*, *S. I.*)

This system was adopted by the *Congrès International pour l'Unification des Filetages*, held at Zurich, October 3–4, 1898. Its proportions differ from those of the French Standard only in the pitches of the screws of 8, 9, 12 and 13 mm. diameter, and in the insertion in the series of the odd numbered diameters 27, 33, 39, 45 mm., which were not included in the French system.

The rules formulated by the Congress apply only to screw-bolts of a nominal diameter of 6 mm. and upward. The form of the thread is practically that of the Sellers system, excepting that a serious defect in the latter is avoided by providing clearance at the bottom of the thread. This clearance must not exceed one sixteenth the height of the primitive triangle. The top of the thread is flat in order to facilitate production and to reduce the liability to injury. The shape of the bottom may be flat or rounded, the latter being recommended to avoid reëntrant angles which aid rupture. The dimensions prescribed by the Congress

* Dimensions given only for bolts at which change of pitch occurs.

are given in Table XVII. The pitch of any size intercalated between those of standard diameters is to be the same as that of the next smaller diameter. The thread, with the full clearance and curved bottom recommended, is shown in Fig. 27. The formulæ are:

$$\text{Altitude, } a, \text{ primitive triangle} = 0.866p;$$

$$d = D - 2 \times \tfrac{13}{16} a = D - 1.407p;$$

$$s \text{ (maximum)} = \frac{D-d}{2} = 0.7035p;$$

$$f = \text{width of flat} = \frac{p}{8};$$

$$\text{Clearance, } C \text{ (max.)} = \frac{a}{16}.$$

TABLE XVII.

INTERNATIONAL STANDARD SCREW-THREADS.*

Nominal Diameter, D.	Pitch, p.	Nominal Diameter, D.	Pitch, p.	Nominal Diameter, D.	Pitch, p.
mm.	mm.	mm.	mm.	mm.	mm.
6	1.00	20	2.5	48	5.0
7	1.00	22	2.5	52	5.0
8	1.25	24	3.0	56	5.5
9	1.25	27	3.0	60	5.5
10	1.50	30	3.5	64	6.0
11	1.50	33	3.5	68	6.0
12	1.75	36	4.0	72	6.5
14	2.00	39	4.0	76	6.5
16	2.00	42	4.5	80	7.0
18	2.50	45	4.5		

21. The British Association Standard Thread.

This thread was taken, with a slight modification, directly from the Swiss system of Professor Thury whose work, for the small screws used in watches and scientific instruments, was similar to that of Sellers and Whitworth for screw-bolts. Thury's investigation was undertaken in 1876 at the instance of the Geneva Society of Arts. His system, like those which preceded it, was formulated from data obtained by measuring the dimensions of many screws accepted as well proportioned. The form of the

* Bulletin Soc. d'Encour., March, 1899.

British Association thread is shown in Fig. 28. It is similar to that of Whitworth. The angle is 47.5°. The formulæ are:

$$p = 0.9^n;$$
$$D = 6p^{\frac{6}{5}};$$
$$s = 0.6p;$$
$$r = \tfrac{2}{11}p.$$

In these equations the quantities are expressed in millimetres. For screws characterized as No. 0, No. 1, No. 2, etc., the index n has the values 0, 1, 2, etc., respectively. The equation for p gives thus a gradually decreasing series, each pitch being 0.9 of its predecessor. The values of the pitches thus obtained, substituted in the equation for D, give the corresponding diameters in millimetres, two significant figures only being taken. Table XVIII. gives the proportions of this system.

Table XVIII.

British Association Standard Thread.

No.	Exact Dimensions, Millimetres.		Approximate Dimensions, Inches.		
	Diameter, Nominal, D.	Pitch, p.	Diameter, Nominal, D.	Pitch, p.	Threads Per Inch, n
0	6.00	1.00	0.236	0.0394	25.4
1	5.30	0.90	0.209	0.0354	28.2
2	4.70	0.81	0.185	0.0319	31.4
3	4.10	0.73	0.161	0.0287	34.8
4	3.60	0.66	0.142	0.0260	38.5
5	3.20	0.59	0.126	0.0232	43.0
6	2.80	0.53	0.110	0.0209	47.9
7	2.50	0.48	0.098	0.0189	52.9
8	2.20	0.43	0.086	0.0169	59.1
9	1.90	0.39	0.075	0.0154	65.1
10	1.70	0.35	0.067	0.0138	72.6
11	1.50	0.31	0.059	0.0122	81.9
12	1.30	0.28	0.051	0.0110	90.7
13	1.20	0.25	0.044	0.0098	101.0
14	1.00	0.23	0.039	0.0091	110.0
15	0.90	0.21	0.035	0.0083	121.0
16	0.79	0.19	0.031	0.0075	134.0
17	0.70	0.17	0.027	0.0067	149.0
18	0.62	0.15	0.024	0.0059	169.0
19	0.54	0.14	0.021	0.0055	181.0
20	0.48	0.12	0.019	0.0047	212.0
21	0.42	0.11	0.017	0.0043	231.0
22	0.37	0.098	0.015	0.0039	259.0
23	0.33	0.089	0.013	0.0035	285.0
24	0.29	0.080	0.011	0.0031	317.0
25	0.25	0.072	0.010	0.0028	353.0

In 1900, a committee of the British Association, appointed to consider modifications in this thread, recommended, as to the screws and nuts from No. 0 to No. 11 inclusive, that the existing proportions remain unchanged, excepting that the top and bottom of the thread be made cylindrical, showing "flats" in section; and that, to provide clearance, the depth of the thread be increased by one tenth of the pitch, thus reducing the effective diameter by one fifth the pitch. Thus, for screw No. 0, the nominal and effective diameter, as modified, would be 6 and 4.6 millimetres respectively, while the corresponding diameters of the nut thus modified would be 6.2 and 4.8 mm.

22. The Square Thread.

The relative advantages and disadvantages of this form of thread have been discussed in §11. It is used for the transmission of power as in screw-jacks, the leading screw of lathes, etc. It is more costly than the triangular thread since it must be cut in the lathe. The proportions have not been standardized. The practice of two prominent companies is given in Tables XIX. and XX. The corners of the thread are slightly rounded and occasionally a small angle is given its sides, although the thread-form is practically square.

Table XIX.

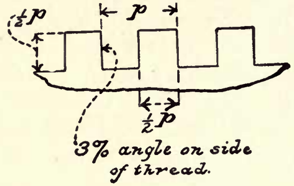
Standard Square Threads.

(William Sellers & Company.)

Nominal Diam., D.	Threads Per In., n.	Effective Diam., d.	Nominal Diam., D.	Threads Per In., n.	Effective Diam., d.
$\frac{1}{4}''$	10	.1625″	$1\frac{3}{8}$	3	1.0834″
$\frac{5}{16}$	9	.2153	$1\frac{1}{2}$	3	1.2084
$\frac{3}{8}$	8	.2658	$1\frac{5}{8}$	$2\frac{3}{4}$	1.307
$\frac{7}{16}$	7	.3125	$1\frac{3}{4}$	$2\frac{1}{2}$	1.4
$\frac{1}{2}$	$6\frac{1}{2}$	.3656	$1\frac{7}{8}$	$2\frac{1}{2}$	1.525
$\frac{9}{16}$	6	.4167	2	$2\frac{1}{4}$	1.612
$\frac{5}{8}$	$5\frac{1}{2}$	.466	$2\frac{1}{4}$	$2\frac{1}{4}$	1.862
$\frac{11}{16}$	5	.512	$2\frac{1}{2}$	2	2.0626
$\frac{3}{4}$	5	.575	$2\frac{3}{4}$	2	2.3126
$\frac{13}{16}$	$4\frac{1}{2}$	.6181	3	$1\frac{3}{4}$	2.5
$\frac{7}{8}$	$4\frac{1}{2}$	.6806	$3\frac{1}{4}$	$1\frac{3}{4}$	2.75
$\frac{15}{16}$	4	.7188	$3\frac{1}{2}$	$1\frac{5}{8}$	2.962
1	4	.7813	$3\frac{3}{4}$	$1\frac{1}{2}$	3.168
$1\frac{1}{8}$	$3\frac{1}{2}$	.875	4	$1\frac{1}{2}$	3.418
$1\frac{1}{4}$	$3\frac{1}{2}$	1			

TABLE XX.

STANDARD SQUARE THREADS.
(NEWPORT NEWS SHIPBUILDING AND DRY DOCK CO.)



Diameter, Nominal, D.	Diameter, Effective, d.	Area, Effective, $\pi d^2 \div 4$.	Threads per In., n.	Nut, Depth, H.
½″	0.3333″	0.0870	6	¾″
⅝	0.4250	0.1418	5	1
¾	0.5500	0.2376	5	1⅛
⅞	0.6530	0.3349	4.5	1¼
1	0.7500	0.4418	4	1½
1⅛	0.8750	0.6013	4	1¾
1¼	0.9640	0.7300	3.5	1⅞
1⅜	1.0900	0.9331	3.5	2
1½	1.1670	1.0700	3	2¼
1⅝	1.2900	1.3070	3	2⅜
1¾	1.4170	1.5770	3	2⅝
1⅞	1.4750	1.7090	2.5	2¾
2	1.6000	2.0106	2.5	3
2¼	1.8500	2.6880	2.5	3⅜
2½	2.0000	3.1416	2	3¾
2¾	2.2500	3.9760	2	4⅛
3	2.5000	4.9087	2	4½

23. The ½-V Screw-Thread.

This thread is a modification of the square and triangular forms designed to combine some of the advantages of both types. The sides are inclined at a moderate angle and there are wide flats at the top and bottom. As compared with the square thread, the various ½-V forms are stronger, being relatively wider at the root; they can be cut with a die, which is not possible with the square thread; the fit in the nut can be made closer; the angularity of the sides facilitates the engagement and disengagement of the divided nuts used with such screws in the lathe and permits also the wear of the thread to be taken up by closing the nut; and finally the thread is cleaned more readily. These advantages are obtained without an excessive increase in friction. It is difficult, however, to keep the cutting tools to the exact angle of the thread.

1. Sellers.—Table XXI. gives the dimensions of the Sellers thread of this type. The formulæ are:

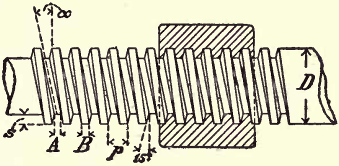
Angle = 15° on side;

Nominal diameter = D.

TABLE XXI.

$\frac{1}{2}$-V SCREW-THREAD.

(WILLIAM SELLERS AND COMPANY.)



Diam. Nominal, D.	Pitch, p.	Threads, No. Per In. n.	Depth of Thread, s.	Width of Flat at Root, A.	Width of Flat at Top, B.	Angle of Thread for Tools.
$\frac{1}{4}$	$\frac{1}{10}$	10	.0438	.038	.0385	7° 15′ 22″
$\frac{3}{8}$	$\frac{1}{8}$	8	.0548	.0475	.0481	6 3 24
$\frac{1}{2}$	$\frac{2}{13}$	$6\frac{1}{2}$	.0674	.0585	.0592	5 35 37
$\frac{9}{16}$	$\frac{1}{6}$	6	.0731	.0633	.0641	5 23 26
$\frac{5}{8}$	$\frac{2}{11}$	$5\frac{1}{2}$	.0797	.0691	.07	5 17 28
$\frac{11}{16}$	$\frac{1}{5}$	5	.0877	.076	.077	5 17 25
$\frac{3}{4}$	$\frac{1}{5}$	5	.0877	.076	.077	4 51 6
$\frac{13}{16}$	$\frac{2}{9}$	$4\frac{1}{2}$	.0974	.0844	.0855	4 58 32
$\frac{7}{8}$	$\frac{2}{9}$	$4\frac{1}{2}$	.0974	.0844	.0855	4 37 18
$\frac{15}{16}$	$\frac{1}{4}$	4	.1096	.095	.0962	4 51 6
1	$\frac{1}{4}$	4	.1096	.095	.0962	4 33 0
$1\frac{1}{8}$	$\frac{2}{7}$	$3\frac{1}{2}$	.1253	.1086	.11	4 37 18
$1\frac{1}{4}$	$\frac{2}{7}$	$3\frac{1}{2}$	.1253	.1086	.11	4 9 40
$1\frac{3}{8}$	$\frac{1}{3}$	3	.1461	.1266	.1283	4 24 45
$1\frac{1}{2}$	$\frac{1}{3}$	3	.1461	.1266	.1283	4 2 46
$1\frac{5}{8}$	$\frac{4}{11}$	$2\frac{3}{4}$	.1594	.1382	.14	4 4 33
$1\frac{3}{4}$	$\frac{2}{5}$	$2\frac{1}{2}$	.1754	.152	.154	4 9 41
$1\frac{7}{8}$	$\frac{2}{5}$	$2\frac{1}{2}$	.1754	.152	.154	3 53 5
2	$\frac{4}{9}$	$2\frac{1}{4}$	.1948	.1688	.171	4 2 46
$2\frac{1}{4}$	$\frac{4}{9}$	$2\frac{1}{4}$	.1948	.1688	.171	3 35 52
$2\frac{1}{2}$	$\frac{1}{2}$	2	.2192	.19	.1924	3 38 33
$2\frac{3}{4}$	$\frac{1}{2}$	2	.2192	.19	.1924	3 18 44
3	$\frac{4}{7}$	$1\frac{3}{4}$	.2505	.2172	.22	3 28 11
$3\frac{1}{4}$	$\frac{4}{7}$	$1\frac{3}{4}$	.2505	.2172	.22	3 12 11
$3\frac{1}{2}$	$\frac{8}{13}$	$1\frac{5}{8}$	.2698	.234	.2368	3 12 11
$3\frac{3}{4}$	$\frac{2}{3}$	$1\frac{1}{2}$	.2923	.2532	.2566	3 14 20
4	$\frac{2}{3}$	$1\frac{1}{2}$	.2923	.2532	.2566	3 2 12
$4\frac{1}{4}$	$\frac{16}{23}$	$1\frac{7}{16}$	.3049	.2643	.268	2 58 57
$4\frac{1}{2}$	$\frac{8}{11}$	$1\frac{3}{8}$	.3188	.2764	.28	2 56 41
$4\frac{3}{4}$	$\frac{16}{21}$	$1\frac{5}{16}$	.3340	.2895	.2933	2 55 22
5	$\frac{4}{5}$	$1\frac{1}{4}$	.3507	.304	.308	2 55 56
$5\frac{1}{4}$	$\frac{4}{5}$	$1\frac{1}{4}$	.3507	.304	.308	2 46 36

$$\text{Pitch, } p = 0.48\sqrt{D + 0.625} - 0.35;$$

$$\text{Depth, } s = 0.4384p;$$

$$\text{Flat at root, } A = 0.38p;$$

$$\text{Flat at top, } B = 0.385p;$$

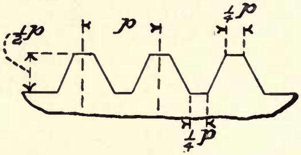
$$\text{Clearance} = 0.01p.$$

2. Newport News Shipbuilding and Dry Dock Company. — The proportions of this thread are given in Table XXII.

Table XXII.

Standard Bastard Screw-Threads.

(Newport News Shipbuilding and Dry Dock Company.)



Diameter, Nominal, D.	Diameter, Effective, d.	Area, Effective, $\pi d^2 \div 4$.	Threads Per In., n.	Width of Flat, $f = \frac{1}{4}p$.	Nut, Depth, H.
$\frac{1}{2}''$	0.3333″	0.0870	6	0.0420	$\frac{5}{8}''$
$\frac{5}{8}$	0.4250	0.1418	5	0.0500	$\frac{7}{8}$
$\frac{3}{4}$	0.5500	0.2376	5	0.0500	1
$\frac{7}{8}$	0.6530	0.3349	4.5	0.0560	$1\frac{1}{8}$
1	0.7500	0.4418	4	0.0625	$1\frac{3}{8}$
$1\frac{1}{8}$	0.8750	0.6013	4	0.0625	$1\frac{1}{2}$
$1\frac{1}{4}$	0.9640	0.7300	3.5	0.0714	$1\frac{5}{8}$
$1\frac{3}{8}$	1.0900	0.9331	3.5	0.0714	$1\frac{3}{4}$
$1\frac{1}{2}$	1.1670	1.0700	3	0.0833	2
$1\frac{5}{8}$	1.2900	1.3070	3	0.0833	$2\frac{1}{8}$
$1\frac{3}{4}$	1.4170	1.5770	3	0.0833	$2\frac{3}{8}$
$1\frac{7}{8}$	1.4750	1.7090	2.5	0.1000	$2\frac{1}{2}$
2	1.6000	2.0106	2.5	0.1000	$2\frac{3}{4}$
$2\frac{1}{4}$	1.8500	2.6880	2.5	0.1000	3
$2\frac{1}{2}$	2.0000	3.1416	2	0.1250	$3\frac{1}{4}$
$2\frac{3}{4}$	2.2500	3.9760	2	0.1250	$3\frac{5}{8}$
3	2.5000	4.9087	2	0.1250	4

3. Acme Standard (29°) Thread. — This form has the same depth as the similar square thread and its sides are at the same inclination as is now adopted generally in cutting worm gears. The formulæ are:

$$\text{Angle of sides} = 14.5° = 29° \text{ included angle};$$

Number of threads per inch $= n$;

Width of flat at top, $B = \frac{0.3707}{n}$;

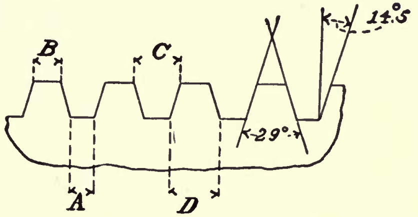
Depth of thread, $s = \frac{1}{2n} + 0.01$;

Nominal diameter $= D$;

Effective diameter $= D - \left(\frac{1}{n} + 0.02\right)$.

TABLE XXIII.

ACME STANDARD (29°) SCREW-THREAD.



No. of Thds. per in. Linear, n.	Depth of Thread, s.	Width at Top of Thread, B.	Width at Bottom of Thread, A.	Space at Top of Thread, C.	Thickness at Root of Thread, D.
1	.5100	.3707	.3655	.6293	.6345
1⅓	.3850	.2780	.2728	.4720	.4772
2	.2600	.1853	.1801	.3147	.3199
3	.1767	.1235	.1183	.2098	.2150
4	.1350	.0927	.0875	.1573	.1625
5	.1100	.0741	.0689	.1259	.1311
6	.0933	.0618	.0566	.1049	.1101
7	.0814	.0529	.0478	.0899	.0951
8	.0725	.0463	.0411	.0787	.0839
9	.0655	.0413	.0361	.0699	.0751
10	.0600	.0371	.0319	.0629	.0681

24. Special Threads.

1. THE KNUCKLE THREAD, Fig. 29, may be considered as formed from the square type by rounding the top and root of the latter in curves which unite. The curvature increases the strength and friction of the thread and reduces its liability to injury in service. Its advantage lies solely in its fitness for rough usage.

2. THE BUTTRESS THREAD, Fig. 30, is a trapezoidal form suitable for producing pressure in one direction only. The driving

side is normal to the axis of the screw as in the square thread the angle between the sides is usually 45° ; and the width of the flat at top and bottom is one eighth of the pitch. For maxi-

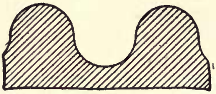


FIG. 29.

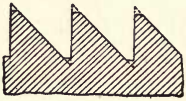


FIG. 30.

mum effort in one direction, the thread has the greatest strength and least friction attainable.

3. MODIFIED BUTTRESS THREAD. — This thread meets extended and important use in the breech-blocks of modern ordnance and also in securing armor-plate to the hulls of war vessels. The profile of armor-threads as used by the U. S. Navy, is shown in Fig. 31, the proportions of various sizes being given in Table

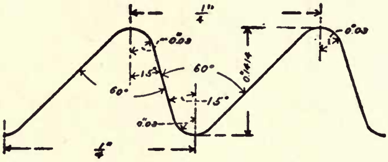


FIG. 31.

XXIV.* One side of the thread makes an angle of 15° with the normal to the axis of the bolt, the similar inclination of the other side is 45°, and the top and root are rounded with ample curves.

TABLE XXIV.

PROPORTIONS OF BOLTS FOR SIDE, DIAGONAL, AND BELT-ARMOR, U. S. NAVY.

Outboard End.		Inboard End.		Thread.			
Diameter, Nominal, D.	Diameter, Effective, d.	Diameter, Nominal, D_1.	Diameter, Effective, d_1.	Depth, s.	Pitch, p.	Radius of Rounding, r.	Armor, Thickness.
2.080″	1.80″	1.780″	1.50″	0.1414″	$\frac{1}{4}$	0.030″	to 5″ inc.
2.680	2.40	2.280	2.00	0.1414	$\frac{1}{4}$	0.030	6 " 9 "
3.200	2.88	2.720	2.40	0.1604	$\frac{2}{7}$	0.035	10 " 13 "
3.680	3.36	3.120	2.80	0.1604	$\frac{2}{7}$	0.035	14 " 17 "
4.216	3.84	3.576	3.20	0.1885	$\frac{1}{3}$	0.040	18 " 21 "

* "Report of Chief of Bureau of Construction and Repair, U. S. Navy," 1896.

4. Modified Triangular Thread. — In modern ordnance, the breech-block is secured by the "Interrupted Screw" method, *i. e.*, the outer surface of the cylindrical block is threaded to form a screw which engages an internal thread in the breech. Neither thread is continuous, each being divided into sectors, 12 in large guns, 6 threaded and 6 blank, the surface of the latter being just below the root of the thread in the former. Each sector is 30° in length and all correspond with similar sectors in the breech-recess of the gun. In closing the breech, the block is placed so that its threaded parts are opposite the blanks of the recess. It is then moved axially home, turned through 30°, and thus locked by the engagement of the threads.

In U. S. naval guns the breech-block thread is of the modified buttress type used upon armor-bolts. Fig. 32 shows in profile

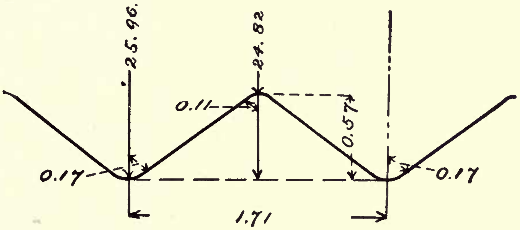


Fig. 32.

the thread of the 16-inch U. S. Army rifle. While the sides of the thread are symmetrical in their inclination to the normal to the axis, the angle between them is large and there is a full rounding at top and root. In this gun the maximum powder-pressure is taken as 37,000 to 38,000 lbs. per sq. in. The dimensions of the block are:

Diameter of breech-block, $D = 25.96''$;
Diameter at root of thread, $d = 24.82$;
Depth of thread, $s = 0.57$;
Pitch of thread, $p = 1.71$;
Radius of rounding, top, $r = 0.17$;
Radius of rounding, bottom, $r_1 = 0.11$;
Length of threaded portion $= 19.89$;
Length of threaded sectors $= 30° - 0.05''$.

25. Machine and Wood Screws.

1. MACHINE SCREWS are those from ½-in. diameter downward, used in metal work. The head is slotted and is either "round" (spherical), "flat" (conical frustum), or "fillister" (cylinder with spherical top). The nominal diameters of these screws are designated by screw-gauge numbers. That of the number 0 screw is 0.05784 in. and the difference between consecutive numbers is 0.01316 in. Therefore, the nominal (outside) diameter (D) of any number (N) may be found from the formula:

$$D = 0.01316\,N + 0.05784.$$

An assortment of pitches is given for each diameter of screw, in order to provide for the use of the same number with either thick or thin pieces, the latter having shallow holes and requiring finer pitches. These screws are described therefore by both the number and pitch. Thus, a "16–18 machine screw" means one of size (screw-gauge number) 16 and 18 threads per inch.

At the meeting of the American Society of Mechanical Engineers, held in May, 1902, Mr. Charles C. Tyler presented a paper on "A Proposed Standard for Machine Screw Sizes." As to present practice, Mr. Tyler states that there are no recognized basic reference standards having a generally accepted form of thread and diameter; that the pitches are apparently standardized only for the sizes having even numbers, although screws and taps are furnished for a number of different pitches for each size; and that the form of the thread varies with different manufacturers. He recommends the adoption of the Sellers form of thread and the computation of the pitch by the formula:

$$p = 0.23\sqrt{D + 0.625} - 0.175,$$

which formula was proposed by Mr. George M. Bond * in 1882, and differs from that of the U. S. Standard only in the coefficient being 0.23 instead of 0.24. The change in this increases the number of threads per inch more rapidly as the diameter decreases. Table XXV., taken from Mr. Tyler's paper, gives present practice and the modifications suggested by him.

2. WOOD-SCREWS. — The maximum diameter of any size of wood-screw is measured by the screw-gauge given in the preced-

* "Standards of Length," G. M. Bond, 1887.

Table XXV.

Machine Screws.

(Present Practice and Suggested Changes.)

Present Diameters and Threads per Inch of Small Machine Screws. The Difference Between Consecutive Sizes is .01316.					Suggested Diameters and Threads per Inch of Small Machine Screws.		
Screw Gauge No.	Stand. No. of Threads per Inch.	Threads also Furnished.	Diam. in Fractional Parts of In	Diameter in Decimal Parts of Inch.	Diameter in Decimal Parts of Inch	Stand. No. of Threads per Inch.	Pitch.
					.050	72	.013889
					.060	64	.015625
1		56,60,64,72.	$\frac{1}{16}$	.07100	.070	60	.016667
$1\frac{1}{2}$		56.	$\frac{5}{64}$	.07758	.080	56	.017857
2	56	48,64.	$\frac{5}{64}$	.08416	.090	52	.019231
3		40,44,48,56.	$\frac{3}{32}$	.09732	.100	48	.020833
4	36	30,32,40,42,44,48.	$\frac{7}{64}$	.11048	.110	44	.022727
5		30,32,36,40,44,48.	$\frac{1}{8}$	.12364	.125	40	.025000
6	32	30,36,38,40,44,48.	$\frac{9}{64}$	.13680	.135	40	.025000
7		24,28,30,32,36,40.	$\frac{5}{32}$	.14996	.150	36	.027778
8	32	24,30,36,40,44.	$\frac{5}{32}$	.16312	.165	32	.031250
9		24,28,30,32.	$\frac{11}{64}$	.17628 }	.180	32	.031250
10	24	20,22,28,30,32,36.	$\frac{3}{16}$	.18944 }			
11		22,24,28,30.	$\frac{13}{64}$	.20260	.200	30	.033333
12	24	20,22,26,28,30,32,34,36.	$\frac{7}{32}$	.21576 }	.220	28	.035714
13		20,22,24,32.	$\frac{15}{64}$	.22892 }			
14	20	16,18,22,24,26.	$\frac{15}{64}$	.24208 }	.250	24	.041667
15		18,20,22,24.	$\frac{1}{4}$	.25524 }			
16	18	16,20,22,24,26.	$\frac{17}{64}$	.26840			
17		16,18,20.	$\frac{9}{32}$	.28156	.28125	22	.045455
18	18	16,20,22,24,26.	$\frac{19}{64}$	.29472			
19		16,18,20,22,24.	$\frac{5}{16}$	.30788 }	.3125	20	.050000
20	16	18,20,22,24.	$\frac{21}{64}$	.32104 }			
22	16	18.	$\frac{11}{32}$	.34736	.34375	20	.050000
24	16	14,18,20,22,24.	$\frac{3}{8}$	.37368	.375	18	.055556
26	16	14.	$\frac{13}{32}$	.40000	.40625	18	.055556
28	14	16.	$\frac{27}{64}$	.42632	.4375	16	.062500
30	14	16.	$\frac{29}{64}$	.45264	.46875	16	.062500
					.500	14	.071429

ing table. These screws differ from those used in metal-work for two reasons: The screw forms its nut as it enters the wood and the material of the nut is much weaker than that of the screw. Therefore, the latter is gimlet-pointed, its body tapers, the threads are thin and sharp, and the space between them is relatively wide in order to provide a wooden nut-thread of sufficient strength.

26. Pipe Threads.

The standard system of pipe-threads now used in the United States was formulated by Mr. Robert Briggs from the average usage of good practice. It was reported upon favorably in 1886

by a committee of the American Society of Mechanical Engineers, was adopted by various associations of manufacturers, and recommended by the American Railway Master Mechanics' Association. The following extract is taken from a paper presented by Mr. Briggs in the Proceedings of the Institution of Civil Engineers of Great Britain, Vol. LXXI.:

"The taper employed for the conical tube-ends is an inclination of 1 in 32 to the axis. . . . A longitudinal section of the tapering tube-end, with the screw-thread as actually formed, is shown in Fig. 33 for a nominal $2\frac{1}{2}$-in. tube, *i. e.*, a tube of about $2\frac{1}{2}$ in. internal diameter and $2\frac{7}{8}$ in. actual external diameter.

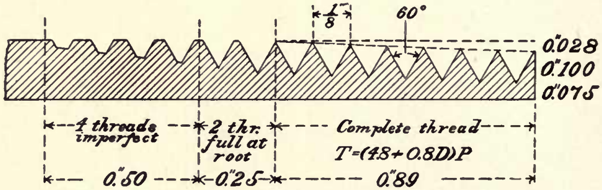


FIG. 33.

"The thread employed has an angle of 60°; it is slightly rounded off both at the top and bottom, so that the height or depth of the thread, instead of being exactly equal to the pitch, is only $\frac{4}{5}$ of the pitch or $0.8/n$, if n be the number of threads per inch. For the length of tube-end throughout which the thread continues perfect, the empirical formula used is:

$$(0.8D + 4.8)/n, \tag{34}$$

where D is the actual external diameter of the tube throughout its parallel length and is expressed in inches.

"Further back, beyond the perfect threads, come two having the same taper at the bottom but imperfect at the top. The remaining imperfect portion of the screw-thread, furthest back from the extremity of the tube, is not essential in any way to this system of joint and its imperfection is simply incidental to the process of cutting the thread at a single operation. From the foregoing, it follows that, at the very extremity of the tube, the diameter at the bottom of the thread is:

$$D - \left[\frac{2(0.8D + 4.8)}{32n} + \frac{2 \times 0.8}{n}\right] = D - (0.05D + 1.9)\frac{1}{n}. \tag{35}$$

"The thickness of iron below the bottom of the thread, at the tube extremity, is taken empirically to be:

$$0.0175D + 0.025. \tag{36}$$

"Hence, the actual internal diameter, d, of any tube is found to be in inches:

$$d = D - (0.05D + 1.9)/n - 2\,(0.0175D + 0.025)$$

$$\therefore\ d = 0.965D - 0.05D/n - 1.9/n - 0.05." \tag{37}$$

The proportions of the Briggs thread are given in Table XXVI. As compared with the Sellers system, the depth of the thread is

measured by a greater fraction of the pitch; but the latter is much finer for a given outside diameter and the thread is therefore shallower and more suitable for the thin walls of a tube.

Table XXVI.

Wrought-Iron Welded Tubes.

(Briggs Standard.)

Taper of Conical Tube End ¾ Inch Per Foot, or 1 in 32 to Axis of Tube.

Diameter of Tube.			Thickness of Metal, Inches.	Screwed Ends.			
Nominal Inside, Inches.	Actual Inside, Inches.	Actual Outside, Inches.		Number of Threads per Inch.	Length of Perfect Thread at Bottom, Inches.	Diameter of Bottom of Thread at End of Pipe, Inches	Diameter of Top of Thread at End of Pipe, Inches.
⅛	0.270	0.405	0.068	27	0.19	0.334	0.393
¼	0.364	0.540	0.088	18	0.29	0.433	0.522
⅜	0.494	0.675	0.091	18	0.30	0.567	0.656
½	0.623	0.840	0.109	14	0.39	0.701	0.815
¾	0.824	1.050	0.113	14	0.40	0.911	1.025
1	1.048	1.315	0.134	11½	0.51	1.144	1.283
1¼	1.380	1.660	0.140	11½	0.54	1.488	1.627
1½	1.610	1.900	0.145	11½	0.55	1.727	1.866
2	2.067	2.375	0.154	11½	0.58	2.200	2.339
2½	2.468	2.875	0.204	8	0.89	2.620	2.820
3	3.067	3.500	0.217	8	0.95	3.241	3.441
3½	3.548	4.000	0.226	8	1.00	3.738	3.938
4	4.026	4.500	0.237	8	1.05	4.235	4.435
4½	4.508	5.000	0.246	8	1.10	4.732	4.932
5	5.045	5.563	0.259	8	1.16	5.291	5.491
6	6.065	6.625	0.280	8	1.26	6.346	6.546
7	7.023	7.625	0.301	8	1.36	7.340	7.540
8	7.982	8.625	0.322	8	1.46	8.334	8.534
9	8.937	9.625*	0.344	8	1.57	9.328	9.528
10	10.019	10.750	0.366	8	1.68	10.445	10.645

27. Stresses in Screw-Bolts.

The body of a screw-bolt may be regarded as a cylindrical bar, subjected in various services either to simple tension or compression or to one of these stresses combined with torsion, or, as in the flanged coupling, to tension and cross-shear. The thread may be considered as a cantilever beam whose section is that cut by a plane passing through the axis, as *O-A*, Fig. 34. The *length* of this assumed beam is the depth of the thread, s; its *depth* at the support is $p - f$, where $p =$ pitch and $f =$ width of flat at root; and its *breadth* at the root is the developed distance through which the axial section *B-C-E* extends. This distance, for one convolution $= \pi d$ and for the threads engaged by a nut of depth H ins. and

* Originally, 9.688.

having $n = 1/p$ threads per inch $= \pi d \times Hn$. Let the total axial load on the bolt $= W$; the load per convolution engaged $= W/Hn = w$; and the permissible tensile and shearing stresses per sq. in. $= S_t$ and $S_s = 0.8S_t$, respectively. Consider the assumed beam with regard to:

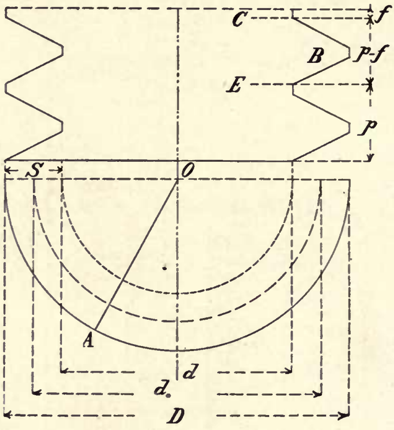


FIG. 34.

1. SHEARING OF THE THREAD, *i. e.*, "stripping" at the root. The shearing force $= W$ and is opposed by the section of metal at the support or root. The area of this section = breadth × depth of beam

$$\therefore \textit{Resistance to Stripping} = W = \pi dHn(p - f)S_s. \tag{38}$$

Equating this, for equal strength throughout, to the tensile resistance of the bolt:

$$\textit{Tensile Resistance} = W = \frac{\pi}{4} \cdot d^2 S_t = \pi dHn(p - f)S_s \tag{39}$$

$$H = \frac{S_t}{S_s} \cdot \frac{d}{4n(p - f)} = \frac{5pd}{16(p - f)}. \tag{40}$$

In the Sellers system, $f = p/8$. Substituting:

$$H = 0.357d.$$

2. RUPTURE OF THREAD by bending at the root. Theoretically, the load is uniformly distributed over the surface, which assump-

tion could be true only of new and perfect threads; practically it may be considered as concentrated at the mean thread diameter. Therefore:

$$\textit{Moment of Load} = W \times \frac{s}{2} = M = S_t I/c;$$

$$\textit{Section-Modulus at Root} = \frac{I}{c} = \frac{Hn\pi d(p-f)^2}{6};$$

$$\textit{Resistance to Rupture} = W = \frac{Hn\pi d(p-f)^2 S_t}{6} \times \frac{2}{s}. \qquad (41)$$

Equating (41) and (39):

$$H = \tfrac{3}{4}\frac{pds}{(p-f)^2}, \qquad (42)$$

which expression assumes the tensile stress in the bolt and that in the thread due to flexure to be of the same intensity. Substituting the values for the Sellers system:

$$H = 0.637d.$$

3. Bearing Pressure upon the Thread. — The allowable pressure upon the area of the engaged threads, as projected upon a plane normal to the axis, depends upon the service of the screw, being much greater with fastenings than with screws for the transmission of power, since, with the latter, friction and wear should be as small as possible. The projected area of the threads engaging the nut is (Fig. 34):

$$\frac{\pi}{4}(D^2 - d^2) \times Hn.$$

Letting $S_b =$ permissible bearing stress per sq. in.:

$$\textit{Permissible Load in Bearing} = W = \frac{\pi}{4}(D^2 - d^2)HnS_b. \qquad (43)$$

Equating (43) and (39):

$$H = \frac{S_t}{S_b} \cdot \frac{pd^2}{D^2 - d^2}. \qquad (44)$$

(*a*) *Fastenings.* — Letting $a =$ effective area of bolt and $A =$ aggregate projected area of engaged threads:

$$aS_t = AS_b \quad \text{and} \quad \frac{S_b}{S_t} = \frac{a}{A}.$$

This stress-ratio, the reciprocal of that in (44), is given for the Sellers system ($H = D$) in Table XXVII.* It will be seen that,

* "Report of the Board to Recommend a Standard Gauge for Bolts, Nuts and Screw-Threads for the U. S. Navy," May, 1868.

in this system, as the nominal diameter increases, there is an increase also in the bearing pressure, the latter varying from 0.242 to 0.331 of the permissible tensile stress per sq. in. Thus, for a 2-in. bolt of metal whose ultimate tensile stress is 60,000 lbs. per sq. in., the permissible tensile stress, allowing for torsion $= S_t = 7,000$ lbs. per sq. in. From the table, $S_b / S_t = 0.3046$, whence $S_b = 2132.2$ lbs. per sq. in., which pressure is about the maximum allowable for fastenings.

TABLE XXVII.

RATIO OF BEARING PRESSURE TO TENSILE STRESS.

(SELLERS SYSTEM.)

Nominal Diameter of Bolt = D.	Effective Area of Bolt = a	Projected Area of Engaged Threads = A.	$\frac{a}{A} = \frac{S_b}{S_t}$	Nominal Diameter of Bolt = D.	Effective Area of Bolt = a.	Projected Area of Engaged Threads = A.	$\frac{a}{A} = \frac{S_b}{S_t}$
$\frac{1}{4}$ in.	.02688	.11105	.242	2 in.	2.3019	7.5573	.3046
$\frac{5}{16}$ "	.04524	.17696	.2556	$2\frac{1}{4}$ "	3.0232	9.6471	.3134
$\frac{3}{8}$ "	.06789	.25536	.266	$2\frac{1}{2}$ "	3.7188	11.8990	.3125
$\frac{7}{16}$ "	.09347	.34826	.2684	$2\frac{3}{4}$ "	4.6224	14.4881	.3190
$\frac{1}{2}$ "	.12566	.45949	.273	3 "	5.4283	17.2221	.3150
$\frac{9}{16}$ "	.16189	.58461	.2769	$3\frac{1}{4}$ "	6.5009	20.4158	.3188
$\frac{5}{8}$ "	.20174	.72222	.2795	$3\frac{1}{2}$ "	7.5477	23.5849	.3200
$\frac{3}{4}$ "	.30190	1.0470	.288	$3\frac{3}{4}$ "	8.6416	27.0337	.3196
$\frac{7}{8}$ "	.41969	1.4303	.293	4 "	9.9929	30.8820	.3235
1 "	.55024	1.8813	.3112	$4\frac{1}{4}$ "	11.328	34.9236	.3244
$1\frac{1}{8}$ "	.69399	2.2877	.3033	$4\frac{1}{2}$ "	12.743	40.3586	.3157
$1\frac{1}{4}$ "	.89082	2.94245	.3027	$4\frac{3}{4}$ "	14.250	43.2728	.3288
$1\frac{3}{8}$ "	1.0568	3.5310	.2993	5 "	15.763	48.4000	.3260
$1\frac{1}{2}$ "	1.2948	4.2507	.3051	$5\frac{1}{4}$ "	17.572	53.4950	.3290
$1\frac{5}{8}$ "	1.5152	4.9925	.3035	$5\frac{1}{2}$ "	19.267	58.6676	.3280
$1\frac{3}{4}$ "	1.7460	5.7750	.3023	$5\frac{3}{4}$ "	21.262	62.7850	.3286
$1\frac{7}{8}$ "	2.0510	6.6572	.3081	6 "	23.098	69.8540	.3310

(*b*) *Screws for Transmitting Power.* — In such screws, the bearing pressure varies within fairly wide limits, being determined by the character and duration of the work. Reuleaux gives 700 pounds per square inch of projected area for square and trapezoidal threads, which pressure is given also by Weisbach for square threads. Unwin states that for screws constantly in motion this pressure should not exceed 200 pounds, and that with no power-screw should it be more than 1,000 pounds.

4. TENSION UNDER STATIC LOAD. — Under this stress, the body of a screw-bolt has a higher elastic limit and a greater ultimate

strength than a cylindrical bar of the same metal and of diameter equal to that at the root of the thread. These gains are due to:

(*a*) *The Reinforcing Action of the Thread.* — When a cylindrical bar is subjected to simple tension only, it is increased in length and contracted in sectional area. The contraction is gradual, extends over a considerable portion of the specimen, and reaches a minimum at the point where rupture occurs finally. To permit the gradual tapering of the specimen in unrestricted contraction, the bar should be originally of the same diameter throughout the section subject to elongation.

If, now, there be turned in the bar one or a series of parallel grooves of any form but of the same depth, the tensile stress and the tendency to elongation and to contraction of area will be greater in the portions of lessened diameter. This reduced section is, however, insufficient in length to permit considerable contraction; and, further, the latter is resisted by the metal under less stress in the ridges of the grooves. In other words, in addition to the lessened distance of least diameter through which stretching occurs, the ridges oppose the contraction of area and the consequent elongation of the reduced section and therefore add to the strength of the latter. As a result, the "grooved specimen" is stronger under static tensile load than a cylindrical bar having the same diameter as that at the base of the grooves.

Mr. Kirkaldy * was the first to emphasize the effect of the form of a specimen upon its ultimate strength. In his report upon the Essen and Yorkshire iron plates, he says:

"When the breadth of a specimen is reduced to a minimum at one point, a greater resistance is offered to its stretching than when formed parallel for some distance; and, as the stretching is checked, so will also be the contraction of area and with it will be an increase in the ultimate stress."

Table XXVIII. gives the results of tests made by Mr. James E. Howard † upon six specimens from the same $1\frac{1}{4}$-in. steel bar, to illustrate the effect of turning a reduced section or "stem," 0.798 in. in diameter on each specimen. Nos. 1, 2 and 3 had cylindrical stems, 1 in., 0.5 in., 0.25 in. long, respectively, connected by full fillets to the body; in specimens Nos. 4 and 5, the stems were semicircular grooves of 0.4 in. and 0.125 in. radius, respectively; a V-shaped groove was formed in specimen No. 6.

* "Experiments on Wrought Iron and Steel," 1862, p. 74.

† "Proceedings International Engineering Congress," 1893.

Table XXVIII.

Grooved Specimens.

No.	Elastic Limit, Pounds Per Sq. In.	Tensile Strength, Pounds Per Sq. In.	Contraction of Area, Per Cent.
1	64,900	94,400	49.0
2	65,320	97,800	43.4
3	68,000	102,420	39.6
4	75,000	116,380	31.6
5	86,000, about.	134,960	23.0
6	90,000 "	117,000	Indeterminate.

In Table XXIX. there are given the results of tests made by Professor Martens which show that a screw-bolt under static tensile load is practically equivalent to a specimen with grooves turned in it of the same form as the thread-groove and also that there was an average increase of 14 per cent. in strength for the specimens tested over that of the cylindrical bar having the same diameter as that of the root of the thread. The table and the following particulars are taken from Professor J. B. Johnson's abstract of Professor Martens' paper:*

"Two grades of mild steel were used for these bolts, all of which were cut from round bars originally 35 mm. (1.4 in.) in diameter. The softer material, having a tensile strength of 53,500 lbs. per sq. in., was used for screw-bolts approximately 1 in. in diameter, and the harder material having a tensile strength of 62,000 lbs. per sq. in. was used for the bolts which were reduced to approximately $\frac{1}{2}$ in. in diameter. Four such bolts were made of each of these sizes of the four styles of thread (sharp V, angle 55°; Whitworth; Sellers, and German Society of Engineers. The latter having an angle of 53° 8′ with flats whose height is one eighth that of the primitive triangle), making in all 32 bolts with screw-threads which were tested. Two of each of these sets were tested in plain tension, the pulling force being applied to the inner face of the nut at one end and increased until rupture occurred.

"The other two bolts of each set were tested also in tension, but under a torsional action resulting from the continuous turning of the nut as the load increased to rupture. In this case the distortion resulting from the permanent elongation of the bolt was nearly all taken up by the movement of the testing machine, the distortion taken up by the turning of the nut being the least possible to maintain a continuous torsional action at this point.

"The same bars were also tested as plain tension-test specimens with cylindrical bodies and again with grooves turned into them of the same shape as the corresponding screw-threads, leaving the same diameter at the bottom of the groove as obtained at the base of the threads."

The ratio $f_{sa} \div f_g$, given in Table XXIX., is practically unity showing that the grooved and threaded specimens are equal in strength. The ratio, $f_{sa} \div$ *test bar*, ranges between 110 and 119 and averages 114, giving thus a mean excess of strength of 14

* *Zeits. d. Ver. Deuts. Ing.*, April 27, 1895. Abstract by Professor Johnson in "Digest of Physical Tests," July, 1896.

Table XXIX.

Threaded and Grooved Specimens.

(Mean Breaking Strength in Pounds per Square Inch.)

Kind of Thread.	Diam. = 1 Inch.									Diam. = ½ Inch.								
	Stress Applied by			Proportion.						Stress Applied by			Proportion.					
	Machine.		Nut.	Test Bar = 100.			$f_g = 100$			Machine.		Nut.	Test Bar = 100.			$f_g = 100$.		
	Grooved.	Threaded.	Threaded.							Grooved.	Threaded.	Threaded.						
	f_g	f_{sa}	f_{sb}	f_g	f_{sa}	f_{sb}	$f\frac{sa}{g}$	$f\frac{sb}{g}$		f_g	f_{sa}	f_{sb}	f_g	f_{sa}	f_{sb}	$f\frac{sa}{g}$	$f\frac{sa}{sb}$	
(*a*) Sharp under 55° . . .	62,430	61,580	49,920	116.8	115.2	93.4	98.9	80.0		71,100	70,400	62,720	114.9	114.0	101.4	99.2	88.2	
(*b*) Whitworth	62,160	61,300	44,800	116.2	114.6	83.8	98.9	72.2		67,000	69,400	58,880	109.4	112.2	95.2	100.3	86.9	
(*d*) Sellers . . .	60,300	60,020	52,330	112.8	112.2	97.9	99.6	86.9		70,250	68,120	62,720	113.6	110.1	94.5	96.9	83.2	
(*c*) German Soc. of Eng'rs.	60,730	61,160	47,640	113.6	114.4	89.1	100.7	78.6		69,260	73,670	62,720	112.0	119.1	101.4	106.4	90.8	
Means				114.9	114.1	91.1	99.5	79.4					112.5	113.9	98.1	100.7	87.3	

Breaking Strength, Normal Test Bar: 1-inch, = 53,480; ½-inch, = 61,860.

per cent. for the threaded rods as compared with cylindrical bars of the same *net* area of cross-section. These results apply only to static or gradually applied loads.

It will be noted that the tensile load upon the cross-section of a bolt at the root of the thread is not uniform throughout, since the metal of the latter opposes the elongation of the section immediately adjacent at the root, thus increasing its stress beyond that existing at the axis. It is apparent that, other things equal, the finer the pitch the more equable will be the distribution of the stress upon the minimum cross-section and the greater the resilience or internal work of the bolt before final yielding. Thus, Major W. R. King, U. S. A., in experimenting with gradually applied loads upon wrought-iron bolts of one and one half inch nominal diameter, U. S. standard, but of varying pitch, obtained results as follows: *

Threads per Inch,	6	12	18
Relative Tensile Strength,	1	1.21	1.23
Elongation,	0.025	0.06	0.08
Relative Internal Work,	1	2.9	4

The U. S. standard pitch for one and one half inch nominal diameter gives 6 threads per inch. The bolts with 18 threads per inch were the stronger. They yielded finally, neither by stripping nor by fracture at the root, but by lateral contraction, so that the threads of bolt and nut became disengaged.

(*b*) *Increased Density of Threaded Section.* — Mr. Kirkaldy † found that, when the thread was cut with new dies, the strength of the threaded section averaged 72.5 per cent. of that of a cylindrical bar whose diameter was that of the outside of the thread. When, however, old and worn dies were used, the average strength was increased to 85 per cent. In the latter case the tendency of the tool is to force aside and compress the metal rather than to remove it by clean cutting, thus increasing the density and strength of the thread and adjacent parts.

Again, in bolts threaded by the "cold-pressed" method, no metal is removed but the thread is raised or spun above the body of the bolt so that the diameter of the shank is intermediate between those of the top and root of the thread. In frequent tests ‡ of mild steel bolts threaded by this method, fracture, under

* *Trans. Am. Inst. Mining Engineers*, 1885.

† Box: "Strength of Materials," 1883, p. 12.

‡ Catalogue Am. Iron and Steel Mfg. Co., 1899.

a gradually applied tensile load, occurred in every instance in the shank, leaving the threaded portion intact. The normal reinforcing action of the thread is, by this process, aided doubtless through the compression and increased density of the thread and adjacent metal.

(*c*) *Résumé.* — The experiments of Professor Martens show that, for static or gradually applied loads, the ultimate strength of the section at the root of the thread is 14 per cent. greater than that of a cylindrical bar of the same metal and cross-section. This increase in strength is due to the reinforcing action of the thread, and, in some degree, to the greater density of the metal. Under sudden and repeated stresses, however, the results would probably be less favorable, owing to the appreciable concentration of the stress about the bottom of the groove which would produce fracture at the reëntrant angle. The increase in strength of the screw from these causes is, therefore, not considered in designing bolts.

5. TENSION UNDER SUDDEN LOADS OR IMPACT. — In both machinery and structures a bolt may be required to withstand not only the tensile stress of a gradually applied or static load but also that produced by a suddenly applied load or by impact. Examples of such requirements may be found in bridge work, in marine machinery, in rock drills, etc.

Let the static or gradually applied load, P, produce in the bolt a total stress, P, and elongation, λ. Then, the same load, if suddenly applied, will produce a maximum, momentary, total stress, $2P$, and elongation, 2λ, which, after a series of axial oscillations of the bolt, will be reduced, when the latter comes to rest, to the final stress, P, and elongation, λ, due to P as a static load. In impact, the load, P, is assumed to act as if it were not only suddenly applied but in motion with a velocity, v, such as would be acquired by fall through a height, h. Under these conditions, P produces a maximum, momentary, total stress, Q, and elongation, y, which, when the bolt after oscillation comes to rest, are reduced to P and λ, respectively. Disregarding the weight and consequent inertia of the bolt, we have:*

$$Q = P\left(1 + \sqrt{2\frac{h}{\lambda} + 1}\right); \tag{45}$$

$$y = \lambda\left(1 + \sqrt{2\frac{h}{\lambda} + 1}\right). \tag{46}$$

* Merriman, "Mechanics of Mechanics," 1900. Art. 93.

When $h = 0$, these formulæ become:

$$Q = 2P \quad \text{and} \quad y = 2\lambda,$$

i. e., the values for a load suddenly applied but without impact.

In the three cases cited, the total final stress is P. For this stress, the absolute requirement is that the area, a, of the minimum cross-section of the bolt shall be such that the unit stress, P/a, shall not exceed the working stress of the metal. The strength of this minimum section is therefore practically the measure of the resistance of the bolt to safe static stress.

Work is the product of a resistance by the distance through which the latter is overcome. The external work of impact, $P(h + y)$, is resisted by the elastic resilience or internal work, $\frac{1}{2}Q \times y$, of the bolt. The same internal work may be the product of a high average, total stress, $\frac{1}{2}Q$, and a small elongation y, or, conversely, of a low stress and a large elongation. Under the conditions given, it is apparent that the elastic resilience is the measure of the resistance of the bolt to sudden or impulsive stress.

In order to secure maximum total elongation under sudden load and therefore the least value of Q, the sectional area of the unthreaded portion of the bolt should be the minimum permissible, *i. e.*, that at the root of the thread, which minimum area is determined by the static load. The minimum section should extend through as great a portion of the bolt as possible, since the total elongation depends upon its length. When the area at any point is greater than the minimum, the unit stress over that area is less than over the latter and the elongation of that part and therefore of the bolt will be reduced proportionately and there will be an increase in the average stress.

Equating the external and internal work, we have for a bar of sectional area A, length L, and maximum total and unit stresses, Q and q, respectively:

$$K = \tfrac{1}{2}\frac{q^2}{E} \cdot AL, \tag{47}$$

on which K is the internal work or elastic resilience and E is the modulus of elasticity for tension.

Consider two bolts of the same total length, length of shank, and area, a, at the root of the thread. In:

Bolt No. 1: Let the length of threaded portion be l and its minimum sectional area and maximum unit stress be a and q, respectively. Let the length of shank be kl and its sectional area be na. Then, the maximum unit stress in the shank will be:

$$\frac{a}{an} \cdot q = \frac{q}{n}.$$

Bolt No. 2: As before, total length $= l + kl = l(1 + k)$. Let the uniform sectional area throughout screw and shank (disregarding thread ridges) $= a$ and the maximum unit stress throughout $= q_1$.

The elastic resilience of each bolt will be the sum of the internal work of its threaded portion and shank. From (47), we have for:

Bolt No. 1:

$$K = \frac{1}{2E}\left(q^2al + \frac{q^2}{n^2} \times na \times kl\right) = \frac{q^2al}{2E}\left(1 + \frac{k}{n}\right) \tag{48}$$

$$q = \sqrt{\frac{2EK}{al} \cdot \frac{n}{n+k}}.$$

Bolt No. 2:

$$K = \frac{q_1^2al}{2E}(1 + k); \tag{49}$$

$$q_1 = \sqrt{\frac{2EK}{al} \cdot \frac{1}{1+k}} = \sqrt{\frac{2EK}{al} \cdot \frac{n}{n+nk}}.$$

Assuming the total work, K, as the same in each case, it will be seen that $q_1 < q$, *i. e.*, that, by making the shank of the same sectional area as that at the root of the thread, the maximum unit stress upon the bolt has been reduced. The equations disregard the increase of area due to the thread ridges, which increase, for accuracy, should be included. When there is no impulsive load and a rigid connection is required, there is no advantage, possibly the reverse, in increasing the elastic resilience of the bolt by decreasing the cross-section of the shank.

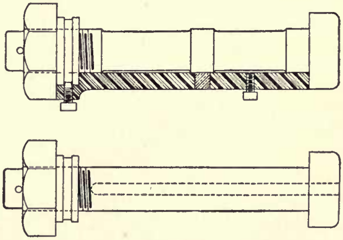


FIG. 35.

In reducing its section, the shank may be turned down on the outside to the diameter at the root of the thread or it may be drilled axially from the head inward to the point where the thread begins, both as in Fig. 35. The latter method is preferable, since it leaves a section which is the stronger of the two in torsion. The shearing stress at any point of a section varies directly as the distance of that point from the axis, but the resisting moment of that stress with respect to the axis varies directly as the square of that distance. Therefore, a given area of section is most economically used with regard to torsion by so disposing it that its fibres shall be remote from the axis.

Professor Sweet,* in testing solid and drilled bolts, $1\frac{1}{4}$ in. nominal diameter and 12 ins. long, found that, under gradually applied load, the undrilled bolt broke in the thread with an elongation of $\frac{1}{4}$ in., while the drilled bolt was fractured in the shank after a total elongation of $2\frac{1}{4}$ ins. Assuming the same mean load in each case, the ultimate resilience of the drilled bolt was 9 times that of the other. "Drop tests," *i. e.*, those producing tensile shock, gave similar results.

6. FRICTION OF THE SCREW.—The screw-thread is essentially an inclined plane wrapped around a cylinder, as on the bolt, or within a hollow cylinder, as in the nut. If the bolt be vertical, the wrench engages the nut in a horizontal plane and the axial load upon the bolt may be assumed as raised vertically by movement along the inclined plane of the nut-thread, the force acting horizontally. The efficiency of the screw, *per se*, and that of the inclined plane are the same. Sliding friction is generated between the bolt and nut threads as they move upon each other. The resistance or force of this friction acts along the contact-surfaces in opposition to the direction of relative motion of the latter. The magnitude of this force is measured by the product of the coefficient of friction and the total normal pressure between the surfaces.

Thread-friction not only reduces the useful work and efficiency of the screw, but also adds to the torsional stress within the body of the bolt produced by the component of the load which is normal to the axis. Therefore, the bolt is subjected, in screwing up, to torsion due to the nut and to tension or compression from the axial load. The combined stress thus developed, exceeds materially the simple axial stress when the nut is screwed home and at

* A. W. Smith, "Machine Design," 1895, p. 135.

rest. This torsional action is of especial importance in small screws, which may readily be sheared by excessive force upon the wrench.

In addition to the friction of the threads, the efficiency of the screw is reduced further by the friction of the rotating member of the pair — the nut or screw, as the case may be — upon its support. Again if, as is usual, the turning moment is applied at one side only and not as a couple, there is a lateral thrust upon the support with a frictional resistance similar to that of a journal.

(*a*) *Torsion due to Thread Friction.* — The pressure upon the

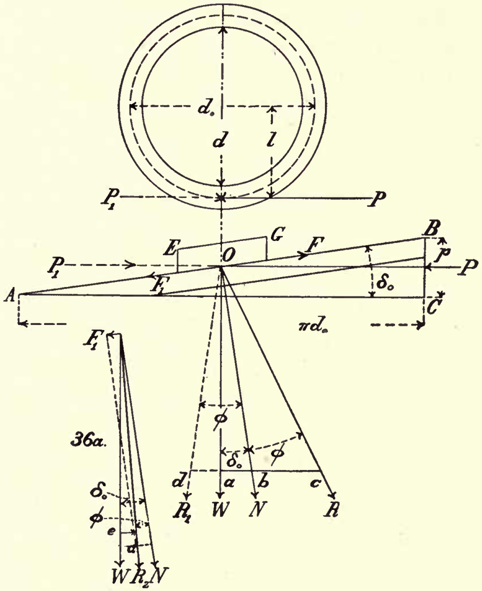


FIG. 36.

threads in computations respecting friction, may be taken as concentrated upon the mean helix or the circumference of the mean thread-diameter, d_0, of pitch-angle, δ_0 (Figs. 22 and 36). Each element of the thread-surface is regarded as sustaining an equal elementary portion of the total axial load or stress, W, and each element has, therefore, a frictional resistance of the same magnitude.

Since the conditions for all elements are thus identical, the total external forces and thread-resistances may be assumed to be concentrated at a single point upon the circumference of diameter, d_0.

In Fig. 36, taking the nut as the turning member, let A-B-C be the inclined plane formed by developing one convolution of the nut thread of diameter, d_0. Let A-B be that thread and E-G a portion of the bolt-thread. The base of the plane is πd_0, the height is the pitch, p, and the pitch-angle, $\delta_0 = B$-A-C. Consider the external rotating force as applied in a plane normal to the axis and as tangent to the mean thread-circumference. Let:

$W =$ total axial load or tension in bolt;

$P_0 =$ external force to raise W without friction;

$P =$ external force to raise W with friction;

$P_1 =$ external force to lower W with friction;

$N =$ direction of thread-pressure, without friction;

$R =$ direction of thread-pressure, raising, with friction;

$R_1 =$ direction of thread-pressure, lowering, with friction;

$\mu =$ coefficient of thread-friction $= \tan\varphi$;

$\varphi =$ angle of repose or of friction;

$F =$ total force of thread-friction in raising $W = N \tan\varphi$;

$F_1 =$ total force of thread-friction in lowering $W = N \tan\varphi$.

Square Threads. — Consider the force upon, and the resistance of, the nut-thread, A-B. To raise W, the latter must move to the left; to lower it, to the right. The resistances of the thread to these movements are the components normal to the axis of N and F and N and F_1 respectively, which resistances must be equal to the corresponding and parallel applied forces, P and P_1.

In raising W without friction:

N is normal to the thread. The resistance is its component normal to the axis and opposing P_0, which component is

$$a\text{-}b = \text{o-}a \tan\delta_0 = W \tan\delta_0 = P_0. \qquad (50)$$

In raising W with friction:

The resistances are the components, normal to the axis, of N and F. The latter $= \mu N = N \tan\varphi$. The resultant of N and F is R, making the angle φ with N. The component of R, normal to the axis and opposing P is

$$a\text{-}c = \text{o-}a \tan(\varphi + \delta_0) = W \tan(\varphi + \delta_0) = P. \qquad (51)$$

From (50) and (51) it will be seen that the resistance of friction is equivalent to increasing the angle B-A-C of a frictionless plane by φ^0.

In lowering W with friction:

The resistance are the components, normal to the axis, of N and F_1. The latter $= \mu N = N \tan \varphi$. The resultant of N and F_1 is R_1, making the angle φ with, and lying to the left of, N. If $\varphi > \delta_0$, the angle between R_1 and the axis is $\varphi - \delta_0$, and the component of R_1 normal to the axis and opposing P_1 is

$$a\text{-}d = \text{o-}a \tan (\varphi - \delta_0) = W \tan (\varphi - \delta_0) = P_1. \qquad (52)$$

In "overhauling" screws, the pitch is so coarse that the load is capable of reversing and lowering the screw. If, in (52), $\delta_0 = \varphi$, then will $\tan (\varphi - \delta_0) = 0 \therefore P_1 = 0$. The pitch-angle is then equal to the angle of repose and no force will be required either to lower the load or to hold it in equilibrium. If, as in Fig. 36, a, $\delta_0 > \varphi$, then R_2, the resultant of N and F_1, will lie to the right of the axis and its component normal to the axis will be e-a, which acts in the direction of the lowering force, P_1. Therefore, the screw, if not sustained by a force, P, will overhaul, with a torque equal to the product of $W \tan (\delta_0 - \varphi)$ by its leverage, $d_0/2$. Screws of this type are met infrequently and, as a rule, in light mechanisms only. Usually, μ lies between 0.10 and 0.20, giving values of φ of about 5° 45′ and 11° 30′, respectively. In Table XIX., for ¼-in. and 4-in. screws, δ_0 is about 8° 45′ and 3° 15′, respectively. These values are for square-threaded, power-screws whose pitch is twice that of corresponding screws of the U. S. Standard.

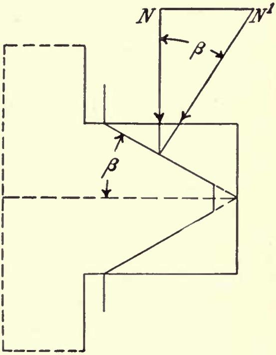


FIG. 37.

Triangular Threads. — In Fig. 37, let N and N' be the normal pressures upon square and triangular threads, respectively. Then $N' = N \sec \beta$, in which β is the base-angle. Letting F' = the frictional resistance of a triangular thread, we have, since for the square thread, $F = \mu N$:

$$F' = \mu N' = (\mu \sec \beta)\, N = F \sec \beta.$$

As compared with the square thread of the same pitch-angle, the friction, F', is, therefore, sec β times greater. Hence, the resisting component, normal to the axis, will be increased proportionately; and, in the formulæ leading to equations (51) and (52), we may replace μ by μ sec β. From these equations, we have:

$$P = W \cdot \frac{\tan \varphi + \tan \delta_0}{1 - \tan \varphi \tan \delta_0} = W \cdot \frac{\mu + \tan \delta_0}{1 - \mu \tan \delta_0} = W \cdot \frac{\mu \pi d_0 + p}{\pi d_0 - \mu p};$$

and, similarly,

$$P_1 = W \cdot \frac{\mu \pi d_0 - p}{\pi d_0 + \mu p}.$$

Replacing μ by μ sec β:

$$P = W \cdot \frac{\mu \sec \beta \pi d_0 + p}{\pi d_0 - \mu \sec \beta p}, \tag{53}$$

$$P_1 = W \cdot \frac{\mu \sec \beta \pi d_0 - p}{\pi d_0 + \mu \sec \beta p}. \tag{54}$$

These are the equations for the raising and lowering forces, P and P_1, respectively, which, considering friction, require to be applied tangentially to the mean thread circumference of a triangular-threaded screw. The form of the equations is that given by Unwin. In the Sellers system, $\beta = 30°$ and sec $\beta = 1.15$. Substituting:

$$P = W \cdot \frac{1.15 \, \mu \pi d_0 + p}{\pi d_0 - 1.15 \, \mu p}, \tag{55}$$

$$P_1 = W \cdot \frac{1.15 \, \mu \pi d_0 - p}{\pi d_0 + 1.15 \, \mu p}. \tag{56}$$

Thus, the $\frac{1}{4}$-in. bolt has, in this system, the maximum inclination of the thread and hence the greatest tendency to be sheared by torsion. For this bolt,

$$p = 0.05 \text{ and } d_0 = \frac{D + d}{2} = \frac{0.25 + 0.185}{2} = 0.2175.$$

Taking $\mu = 0.124$:

$$P = 0.22W.$$

With an ultimate tensile strength of bolt-metal of 60,000 lbs.:

$$W = \frac{\pi d^2}{4} \times 60{,}000 = 1{,}613 \text{ lbs.};$$

$$P = 1613 \times 0.22 = 355 \text{ lbs.},$$

i. e., a force of 355 lbs. applied, under the conditions as above, to a $\frac{1}{4}$-in. screw-fastening will rupture the latter by tensile stress.

The assumed value of μ is suitable only for accurately fitting, well-lubricated threads. Owing to viscidity of the lubricant, the presence of foreign matter, or rough surfaces from abrasion, the coefficient will be usually much higher with a corresponding increase in friction and torsional stress.

(*b*) *Coefficients of Friction for Screw-Threads.* — In average cases, the value of μ is taken as 0.15. This assumes fair conditions of surface and lubrication. Under other circumstances the coefficient may reach 0.40 or more. Professor Albert Kingsbury * has contributed to the meager knowledge available upon this question, the results of valuable experiments conducted by him and applying especially to slow-moving power-screws.

The tests were made upon a set of square-threaded screws and nuts of materials as given in Table XXX. and of dimensions as follows:

Outside Diameter of Screw	1.426	inches.
Inside Diameter of Nut	1.278	"
"Mean Diameter" of Thread	1.352	"
Pitch of Thread	.333	"
Depth (effective) of Nut	1.062	"

The nuts fitted the screws very loosely, so that all friction was excluded except that on the faces of the threads directly supporting the load. Four sets of tests were made. The maximum total load was 14,000 pounds in all tests excepting No. 4, in which it was 4,000 pounds. Readings were taken at pressures given in the table. The total bearing area of thread was approximately one square inch, so that the total axial load was equal to the pressure per square inch upon the thread.

The lubricants were a purely mineral "Heavy Machinery Oil" of specific gravity, 0.912, and "Winter Lard Oil" of sp. gr., 0.919. The former, in test No. 3, was mixed, in equal volumes, with graphite, the brand being Dixon's "Perfect Lubricator." The screws and nuts were flooded with lubricant immediately before the tests.

The threads were carefully cut in the lathe and had been worn down to good condition by previous trials. Screw No. 5 was not quite so smooth as the others. The speed was very slow, being about one revolution in two minutes and the motion, in tightening especially, was also somewhat irregular, so that the action between

* *Trans. Am. Soc. Mech. Engs.*, Vol. XVII.

screw and nut was quite similar to that occurring when machine-bolts are set up in comparatively unyielding material. The results are given in Table XXX. Each figure in test No. 1 is the average of eight readings; in the remaining tests, of four readings.

TABLE XXX.

COEFFICIENTS OF FRICTION FOR SQUARE THREADS.

Screws.	Nuts.				
	6 Mild Steel.	7 Wrought Iron.	8 Cast Iron.	9 Cast Brass.	
1. Mild Steel.	0.141	0.16	0.136	0.136	TEST NO. 1.
2. Wrought Iron.	0.139	0.14	0.138	0.147	Heavy Machinery
3. Cast Iron.	0.125	0.139	0.119	0.171	oil.
4. Cast Bronze.	0.124	0.135	0.172	0.132	Pressure, 10,000
5. Mild Steel, Case Hardened.	0.133	0.143	0.13	0.193	lbs. per sq. in.
1.	0.12	0.105	0.10	0.11	TEST NO. 2.
2.	0.1125	0.1075	0.10	0.12	Lard oil.
3.	0.10	0.10	0.095	0.11	Pressure, 10,000
4.	0.115	0.10	0.11	0.1325	lbs. per sq. in.
5.	0.1175	0.0975	0.105	0.1375	
1.	0.111	0.0675	0.065	0.04	TEST NO. 3.
2.	0.089	0.07	0.075	0.055	Heavy Mach'y oil
3.	0.1075	0.071	0.105	0.059	and Graphite.
4.	0.071	0.045	0.044	0.036	Pressure, 10,000
5.	0.1275	0.055	0.07	0.035	lbs. per sq. in.
1.	0.147	0.156	0.132	0.127	TEST NO. 4.
2.	0.15	0.16	0.15	0.117	Heavy Machinery
3.	0.15	0.157	0.14	0.12	oil.
4.	0.127	0.13	0.13	0.14	Pressure, 3,000
5.	0.155	0.1775	0.1675	0.1325	lbs. per sq. in.

Professor Kingsbury's conclusions are:

"That, for metallic screws in good condition, turning at extremely slow speeds, under any pressure up to 14,000 lbs. per square inch of bearing surface and freely lubricated before application of the pressure, the following coefficients of friction may be used:

COEFFICIENTS OF FRICTION.

Lubricant.	Minimum.	Maximum.	Mean.
Lard Oil,	0.09	0.25	0.11
Heavy Machinery Oil (Mineral),	0.11	0.19	0.143
" " " and graphite in equal volumes,	0.03	0.15	0.07

With regard to the value of the coefficient to be used in designing power-screws, Professor Kingsbury says:

"That (the value) depends upon the object of the design. If the screw is to be made so that it could not overhaul under the most favorable conditions, with either lard oil or

heavy machinery oil, probably 8 per cent. would be the highest allowable coefficient; and, for a certain margin of safety, a somewhat lower figure. If the driving mechanism is to be designed with a view to making the screw turn, even if perfectly dry, probably 30 or 40 per cent. would be the figure. If the amount of power likely to be lost in the long run is what is wanted, probably 15 per cent. would be a safe coefficient for everyday work. This might be reduced to 10 per cent. with lard oil under the best conditions and at the speeds used in these experiments."

Mr. Wilfred Lewis states that, "for feed screws which turn slowly, $\mu = 0.15$ may be taken as a good general average."

(*c*) *Friction of the Support.*—The thrust of a power-screw may be taken by the end of the screw itself upon a plane step-bearing whose maximum diameter is equal to the effective diameter, d, of the screw or the thrust may be borne by an annulus forming a collar-bearing at the end of the threaded portion. Both types of bearing are indicated in Fig. 38. In fastenings, the thrust and force of friction act between the under surface of the nut and the washer, the leverage of the force being about two thirds the nominal diameter, D, of the bolt. Let:

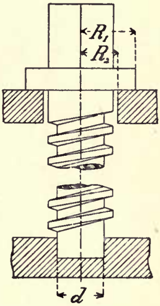


FIG. 38.

W = total axial load;

μ' = coefficient of friction;

$W\mu'$ = force of friction;

r = radius of plane step bearing of diameter, d;

R_1 and R_2 = outer and inner radii, respectively, of collar-bearing;

$R = \frac{2}{3} D$ = leverage of $W\mu'$ in nut.

Then, the moment of the friction in the:

$$\textit{Step Bearing} = W\mu' \cdot \tfrac{2}{3} r; \tag{57}$$

$$\textit{Collar Bearing} = W\mu' \cdot \tfrac{2}{3} \cdot \frac{R_1^3 - R_2^3}{R_1^2 - R_2^2}; \tag{58}$$

$$\textit{Nut} = W\mu' \cdot \tfrac{2}{3} D. \tag{59}$$

The reduction of the moment by the use of a step-bearing is apparent. This form, however, produces the most uneven wear and usually the greatest unit pressure.

In addition to the vertical load there is usually a sidewise thrust on the screw-support, since the power is generally applied as a single force and not as a couple. This produces lateral pres-

sure and friction between the threads or shank of the screw and the support or nut and connected parts. The action resembles that of a shaft journal. Views as to the distribution of friction in the latter are somewhat conflicting. In practice, the total pressure is assumed to be divided uniformly over the projected area of the bearing surface.

7. COMBINED TORSIONAL AND TENSILE OR COMPRESSIVE STRESSES. — The axial load upon a screw produces a tensile or compressive stress and the external force applied to the nut in order to raise the load, develops a shearing stress. Disregarding the reinforcing action of the thread, both stresses may be assumed as acting upon the effective area only of the bolt. Then, the unit tensile stress will be equal to the total load divided by the effective area and the unit shearing stress at the outer circumference of the area — where that stress is a maximum — will be equal to the twisting moment divided by the polar modulus of the section. Referring to Fig. 36, the twisting moment is $P \times d_0/2$. Then:

$$\textit{Unit tensile stress} = W \div \frac{\pi d^2}{4} = S_t; \tag{60}$$

$$\textit{Unit shearing stress} = P\frac{d_0}{2} \cdot \frac{16}{\pi d^3} = \frac{8Pd_0}{\pi d^3} = S_s. \tag{61}$$

These stresses coexist and combine to produce a maximum, unit tensile stress upon a plane whose angle with the axis depends upon their relative magnitude. Similarly, they combine to produce a maximum unit shearing stress upon a plane whose angle differs from that of the first but is governed by similar conditions. Evidently, the required effective area will depend upon the intensity of these resultant stresses, the formulæ for which are:

$$\textit{Maximum tensile unit stress} = \tfrac{1}{2} S_t + \sqrt{S_s^2 + \tfrac{1}{4}S_t^2} = S_t\,max.; \tag{62}$$

$$\textit{Maximum shearing unit stress} = \sqrt{S_s^2 + \tfrac{1}{4}S_t^2} = S_s\;max. \tag{63}$$

When a screw which is so short that it may be treated as a strut, is under compression, the maximum compressive and shearing unit stresses may be found by replacing S_t in (62) and (63) by the unit compressive stress. In designing a screw for a given load, the maximum stresses, as above, must not exceed the elastic strength of the metal. The usual practice, as given in § 29, is to assign a reduced working stress to the material as the diameter decreases.

The experiments of Professor Martens — the results of which are given in Table XXIX. — show the weakening of the effective section of the bolt to axial tensile load which results from the torsional action of the nut. His conclusions, from these tests, are :

"The weakening effect of the turning of the nut under stress at rupture, is much less than might have been predicted, when the distortion of the screw below the nut by permanent elongation is taken into consideration. The tests indicate, for this case, a strength of the 1-in. bolts about 20 per cent. less than that of the plain bars and of the ½-in. bolts about 15 per cent. less than that of the plain bars. In general, it may be said that the turning of the nut upon the bolt at rupture reduces the strength of the nut section of the bolt by about 30 per cent."

8. Cross Shear. — In the flange coupling shown by Fig. 39, the bolts transmit the torsional stress from one section of the shaft to the next, and, if accurately fitted to the bolt-holes, are exposed practically to cross shear only, there being no bending stress and the tensile load, due to drawing the flanges together, being relatively slight. The usual method of design is to assume the diameter of the bolt circle and equate the resistances to shearing of the shaft and bolts, the result being an equation in terms of the diameter and number of the latter. Let :

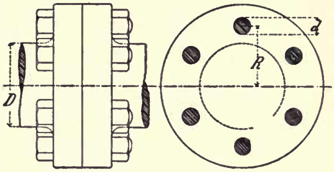


Fig. 39.

R = radius of centre of bolt-holes ;

D = diameter of shaft ;

d = diameter of bolts ;

n = number of bolts ;

$T.M.$ = maximum twisting moment on shaft ; (force, $T.F.$)

$R.M.$ = resisting moment of shaft ;

$T.'M.'$ = twisting moment at bolt centres ; (force, $T.'F.'$)

$R.S.$ = aggregate resistance of bolts to shearing.

The resisting moment to shearing of a circular section is equal to the product of the shearing stress, S_s, at its periphery by the polar modulus of the section, $\pi d^3/16$, where d is the diameter. $T.F.$ is expressed in terms of the unit radius and will be to $T.'F.'$ inversely as their respective radii. We have :

$$T.M. = R.M. = \frac{\pi D^3}{16} \cdot S_s;$$

$$T.'F' : T.F. :: 1 : R \therefore T.'F.' = \frac{T.M.}{R} = \frac{\pi D^3}{16R} \cdot S_s;$$

$$R.S. = \frac{\pi d^2}{4} \times n \times S_s.$$

Equating the values of $T.'F.'$ and $R.S.$:

$$d = \sqrt{\frac{D^3}{4nR}}.$$

To allow for inaccurate fitting and, therefore, for slight bending, the shearing stress on the bolts is usually made three fourths of that on the shaft. Introducing this fraction:

$$d = \sqrt{\frac{D^3}{3nR}}. \qquad (64)$$

R is usually 0.75 to 0.8 times D. The number and diameter of the bolts are interdependent. If it be desired that the outside diameter of the coupling shall be as small as possible, n should be increased and d decreased. n is usually a multiple of the number of duplicate sections of the crank-shaft. The bolts may be either headless taper bolts or "body-bound" and cylindrical with heads, as shown in Fig. 39. With the former type the weight of the head is saved and a rigid joint ensured. The objections to it are the accurate fit required, and, owing to the tapering hole, the impossibility of making the sections of a crank-shaft interchangeable. It will be noted that the analysis assumes the shearing stress to be distributed uniformly over the cross section of the bolt. While this assumption has sufficient practical accuracy, the stress upon the bolt-section varies in intensity, being greatest upon that side of the section which is most remote from the centre of the shaft.

9. Stress in Cylinder-head Studs. — The stress in bolts used in securing steam-cylinder covers and in other joints requiring to be tight against fluid pressure, is affected by somewhat complex conditions. The joint may be made metal to metal and ground or a gasket may be interposed between the flanges. The material of the latter depends upon the steam pressure and the corresponding temperature. Rubber and sheet asbestos, plain or in combination, and copper in corrugated sheets, wire, or wire-gauze, are used for this purpose.

The bolts, the flanges, and the gasket (if any), are all more or less elastic. The bolts are set up with an initial tension which is opposed by the force due to the compression of the flanges and gasket. Later, steam is admitted to the cylinder placing an additional tensile load upon the bolts, which load elongates the latter still further and thus reduces the compressive force, as above. Referring to Fig. 40, let:

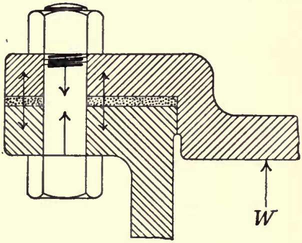


FIG. 40.

S_t = initial unit stress in bolt;

S_c = initial unit force on bolt due to compression of gasket and flanges.

Then: $S_t = S_c$.

When the steam enters the cylinder, the forces acting unon the bolts are the maximum load, W, due to the steam and the reduced compressive force between the flanges. These forces are opposed by the tensile stress within the bolt. Let:

S_w = unit force on bolt corresponding with external load, W;

S_c' = unit force on bolt corresponding with reduced compression between flanges;

S_t' = unit tensile stress in bolt when load, W, is applied.

Then: $S_t' = S_w + S_c'$.

If the bolt stretches by an amount equal to the initial compression of the other members, $S_t' = S_w$, and the joint will open. On the other hand, with a short, rigid bolt, connecting ground flanges without gasket, the elongation will be relatively small and, with high initial stress, the value of S_t' approaches S_w, plus the initial compressive force, S_c. In any event, for a tight joint, the intensity of S_t' must exceed S_w and S_c' must be greater than zero. In Table XXXI., there are given the numbers, diameters, working stresses, and ultimate unit strengths of the cylinder-head studs for the high-pressure cylinders of some of the later vessels of the U. S. Navy. The area under load includes that of the cylinder and counterbore plus, in some cases, a portion of that over the ports. When a cylinder liner is used, the counterbore may be only $\frac{1}{8}$ inch deep.

TABLE XXXI.

STEEL STUDS FOR CYLINDER COVERS. U. S. NAVY.

H. P. Cylinder.				Studs.			
Diameter, Ins.	Initial Press. Gauge, lbs. per sq. in.	Total Area Inside of Flange, sq. ins.	Total Load at Initial Press., lbs.	Number.	Diameter.	Stress per sq. in. of Eff. Area at Initial Press., lbs.	Material of Tensile Strength (Minimum) lbs. per sq. in.
14	250	153	38,250	18	$\frac{3}{4}$	7036	80,000
$20\frac{1}{2}$	250	342.25	85,562	28	$\frac{7}{8}$	7240	80,000
30	200	921.3	184,260	24	$1\frac{3}{8}$	7264	80,000
35	250	1484	371,000	38	$1\frac{1}{2}$	7539	75,000
$38\frac{1}{2}$	250	1720	430,000	38	$1\frac{5}{8}$	7469	75,000

28. Stresses in Nuts.

1. SHEARING, RUPTURE, AND BEARING PRESSURE upon the thread. The conditions as to these stresses are similar to those which exist with the bolt-thread, excepting that, as the diameter at the root of the nut-thread is the nominal diameter, D, plus the

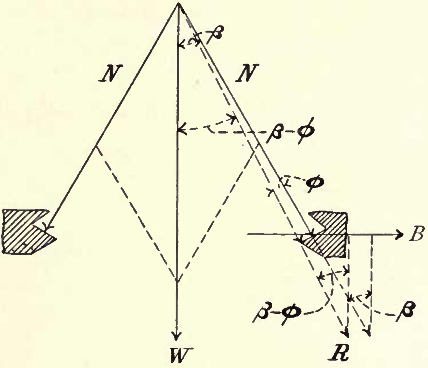


FIG. 41.

clearance spaces, the total section at the root to resist shearing and rupture and the projected area of the thread are slightly greater than those of the bolt.

2. BURSTING STRESS. — In Fig. 41,* let:

W = axial load upon the bolt;

* "Report of Board to Recommend a Standard Gauge for Bolts, Nuts, and Screw-Threads, U. S. Navy," May, 1868.

$N =$ normal pressure upon one half the thread, resolved in a direction perpendicular to any single element of its helicoidal surface;

$B =$ component of N acting in a direction perpendicular to the axis of bolt;

$\beta =$ base-angle of thread;

$\varphi =$ angle of repose or friction.

Then, without friction:

$$W = 2N \cos \beta \therefore N = \frac{W}{2 \cos \beta};$$

$$B = N \sin \beta = \frac{W}{2} \cdot \frac{\sin \beta}{\cos \beta} = \frac{W}{2} \cdot \tan \beta.$$

Considering friction, the true direction of pressure, R, is inclined to the normal, N, by the angle φ; and, as the tendency of W to resist and reverse the nut is opposed by the friction, the bursting effect of W will be reduced and the angle between W and R becomes $\beta - \varphi$. Then:

$$B = \frac{W}{2} \tan (\beta - \varphi). \tag{65}$$

In the Sellers system $\beta = 30°$. Taking $\mu = \tan \varphi = 0.124$, $\varphi = 7° 04'$ and $\beta - \varphi = 22° 56'$. Hence,

$$B = \frac{W}{2} \tan 22° 56' = 0.2115 W.$$

A given axial load, W, produces, then, a bursting pressure, $B = 0.2 W$; and, therefore, the vertical section, through the short diameter of the nut — the stress upon which section resists B — should be two tenths the effective bolt-area, since the stress upon the latter sustains W.

The total width of the resisting section of the nut is $d_n - D$, where $d_n =$ short diameter of nut and $D =$ nominal diameter of bolt; the height of the section is that, $H = D$, of the nut. For convenience, assume the nominal area of the bolt as effective in sustaining W. Then

$$H(d_n - D) = D(d_n - D) = \frac{\pi}{4} \cdot D^2 \times 0.2115,$$

$$\therefore d_n = 1.166D.$$

Since, in this system, $d_n = 1.5D + \frac{1}{16}$ in. for finished nuts, there is a considerable excess of strength to resist bursting.

29. Efficiency of the Screw.

Consider the screw with regard to :

1. LOSS OF POWER.— The efficiency is the ratio between the useful and total work. Disregard journal friction, as absent or uncertain.

(*a*) *Square Threads.* — From Fig. 36 and equations (50) and (51), we have, *for the thread only*, per revolution, in raising W with friction :

$$\frac{\textit{Useful Work}}{\textit{Total Work}} = \frac{P_0.\pi d_0}{P.\pi d_0} = \frac{W \tan \delta_0.\pi d_0}{W \tan (\delta_0 + \varphi).\pi d_0}$$
$$= \frac{\tan \delta_0}{\tan (\delta_0 + \varphi)} = E, \qquad (66)$$

which expression gives the efficiency, E, of the screw-thread for any given pitch-angle, δ_0, of the mean helix and any angle of repose, φ. When the screw is employed solely for transmitting power, the pitch-angle of maximum efficiency should be used, if practical considerations do not prevent. Differentiating (66) and putting the first derivative equal to zero :

$$\frac{dE}{d\delta_0} = \frac{\cot (\delta_0 + \varphi)}{\cos^2 \delta_0} - \frac{\tan \delta_0}{\sin^2 (\delta_0 + \varphi)} = 0;$$

whence $\delta_0 = 45° - \varphi / 2$, which value of δ_0 will make E a maximum. Substituting in (66) :

$$E\,(max.) = \frac{\tan\left(45° - \frac{\varphi}{2}\right)}{\tan\left(45° + \frac{\varphi}{2}\right)}.$$

If $\mu = 0.105$, $\varphi = 6°$, $\delta_0 = 42°$, and $E = 0.81$. Good practical reasons make it undesirable to use so large an angle. Multiple threaded screws, however, owing to their ample bearing surfaces, permit relatively steep pitches.

For the friction of the support, we have, for a screw whose thrust-collar has a mean friction-diameter, D', a work of collar-friction per revolution equal to the force of friction multiplied by its circumferential path $= W\mu'.\pi D' = W \tan \varphi'.\pi D'$. This work

must be added to that expended on the thread in order to find the total work. Hence, *including thread and collar-friction:*

$$E' = \frac{W \tan \delta_0 . \pi d_0}{W \tan (\delta_0 + \varphi) \pi d_0 + W \tan \varphi' . \pi D'}$$

$$= \frac{\tan \delta_0}{\tan (\delta_0 + \varphi) + \frac{D'}{d_0} \tan \varphi'} . \qquad (67)$$

Assuming the same coefficient of friction for thread and collar, $\tan \varphi' = \tan \varphi = \mu$ and E' becomes a maximum when

$$\cot \delta_0 = \mu + \sqrt{\frac{1 + \mu^2}{1 + \frac{D'}{d_0}}} .$$

In the table relating to square-threaded screws which follows, the efficiencies have been calculated, but, in several cases, they have been checked by experiment and found to be fair average values. The efficiency of any screw will, of course, vary widely with the amount of lubrication. The same coefficient of friction —$\mu = 0.15$, $\varphi = 8° 30'$—is taken for both thread and thrust-collar. The diameter of the latter is assumed to be that of the thread. $E =$ the efficiency per cent. when there is no friction between the thrust-collar and its bearing; $E' =$ the efficiency per cent., allowing for thrust-collar friction.

TABLE XXXII.*

APPROXIMATE EFFICIENCIES OF SQUARE THREADED SCREWS.

Angle of Thread, δ_0	E	E
2°	19	11
3	26	14
4	32	17
5	36	21
10	55	36
20	67	48
$45° - \frac{\phi}{2}$	79	52

The efficiency of a square-threaded screw in lowering W may be found from the values of the useful and total work by a process similar to that given for the efficiency in raising the weight.

* Goodman: "Mechanics Applied to Engineering," 1899, p. 204.

(*b*) *Triangular Threads.* — From equation (66), we have for the efficiency of a square thread:

$$\tan \delta_0 \div \frac{\tan \delta_0 + \tan \varphi}{1 - \tan \delta_0 \tan \varphi}.$$

Replacing $\tan \varphi$ by $\mu \sec \beta$, we have, in raising W with friction in triangular-threaded screws, *for the thread only*, the efficiency:

$$E = \tan \delta_0 \cdot \frac{1 - \mu \sec \beta \tan \delta_0}{\tan \delta_0 + \mu \sec \beta}. \tag{68}$$

For the friction of the nut on its washer or boss — assuming the mean friction diameter of the nut as $\frac{4}{3}$ of D, the nominal diameter of the bolt — we have a work of nut-friction per revolution of $W \tan \varphi' \cdot \frac{4}{3}\pi D$, which work must be added to that expended on the thread. Hence, including thread and nut friction, as in (67):

$$E' = \frac{W \tan \delta_0 \cdot \pi d_0}{W \tan (\delta_0 + \varphi)\pi d_0 + W \tan \varphi' \cdot \frac{4}{3}\pi D};$$

$$= \frac{\tan \delta_0}{\dfrac{\tan \delta_0 + \tan \varphi}{1 - \tan \delta_0 \tan \varphi} + \frac{4}{3} \cdot D/d_0 \cdot \tan \varphi'}. \tag{69}$$

Replacing $\tan \varphi$ by $\mu \sec \beta$ and $\tan \varphi'$ by μ':

$$E' = \frac{\tan \delta_0}{\dfrac{\tan \delta_0 + \mu \sec \beta}{1 - \tan \delta_0\, \mu \sec \beta} + \frac{4}{3} \cdot D/d_0 \cdot \mu'}.$$

In the Sellers system, $\sec \beta = 1.15$. Hence:

$$E = \tan \delta_0 \cdot \frac{1 - 1.15\, \mu \tan \delta_0}{\tan \delta_0 + 1.15\, \mu}; \tag{70}$$

$$E' = \frac{\tan \delta_0}{\dfrac{\tan \delta_0 + 1.15\, \mu}{1 - 1.15\, \mu \tan \delta_0} + 1.33\, \mu' D/d_0}. \tag{71}$$

The efficiency of a triangular-threaded screw in lowering W may be found by a similar process.

Mr. Wilfred Lewis gives the following approximate formulæ for the external force and efficiency of triangular-threaded screws, which formulæ he states are applicable with a close degree of accuracy to most of the cases which occur in practice. Let:

p = pitch of screw;
D = outside diameter of screw;
P = force applied at circumference (of screw) to lift a unit of weight;
E' = efficiency of screw in lifting.

Then: $$P = \frac{p + D}{3D} \quad \text{and} \quad E' = \frac{p}{p + D}. \qquad (72)$$

Experiments,* conducted by Mr. James McBride to determine the efficiency of a screw, gave results in accord with the formulæ given above. The test was made with an ordinary 2-inch screw-bolt, not especially prepared. The thread was of the standard V-shape and of 0.22-inch pitch. The nut was not faced and had the flat side to the washer, the latter being of malleable iron, not faced. The contact-surfaces of nut and washer and the threads of nut and bolt were well lubricated with lard oil. The axial, tensile load upon the bolt was 7,500 lbs. The nut was a good fit, and, when not loaded, was easily run up and down the bolt with the fingers. Wrenches of different lengths were applied to the nut and a known force which would just move the latter, exerted upon each wrench. The ratio between the useful work of lifting the weight and the total work expended upon the nut, gave the efficiency, which, for 5 tests, averaged 10.19 per cent.

The effective diameter of a 2-inch bolt = 1.712 in. The mean thread diameter, d_0, is therefore 1.856 in. The pitch = 0.2222 in. and $\tan \delta_0 = p/\pi d_0 = 0.038$ in. Assuming $\mu = 0.15$ and $\mu' = 0.10$, and substituting in formula (71), we find $E' = 10.7$ per cent. Again, substituting the values of p and D in formula (72), we find $E' = 10$ per cent. The theoretical and experimental results are hence practically the same.

2. Loss of Axial Strength. — In screwing up a nut, the bolt is subjected to the tensile or compressive stress corresponding with the axial load produced and to the torsional stress developed through the action of the nut-thread on the bolt-thread. The torsional action results from thread-friction and from that component of the axial load which must be overcome in order to move the latter up the inclined plane of the screw.

The measure of torsion is the twisting moment, *T.M.*, the latter being the product of the force, P, Fig. 36, by its lever-arm $d_0/2$

* *Trans. Am. Soc. Mech. Engrs.*, Vol. XII.

$= l$. For equilibrium, the twisting moment must be equal to the resisting moment. The latter, for a circular section, is the product of the unit shearing stress S_s, at the periphery of the section by the polar modulus of the section, which is $\pi d^3/16$, where d is the diameter. Taking, for convenience, $d_0 = d$, we have $l = d/2$ and:

$$T.M. = Pl = S_s \cdot \frac{\pi d^3}{16} \therefore P = S_s \cdot \frac{\pi d^2}{8}, \qquad (73)$$

i. e., if S_s be the greatest allowable shearing stress in all bolts, the turning force P, which may be applied as above with safety, varies as the square of the diameter. This condition prevails also with the axial load, since that load by (60) is

$$W = \frac{\pi d^2}{4} \cdot S_t,$$

in which S_t is the greatest allowable tensile unit stress.

The relation between the twisting force and the axial load is given by (51) as:

$$P = W \tan(\delta_0 + \varphi).$$

φ is here a constant for all screws under similar conditions of surface and lubrication. The angle, δ_0, is, however, in the Sellers system, variable, being a maximum at the smallest diameter. For example, it is $4^\circ\ 11'$ for the $\frac{1}{4}$-inch screw and $1^\circ\ 45'$ for the 3-inch screw. Replacing W in (51) by its equivalent:

$$P = S_t \cdot \frac{\pi d^2}{4} \cdot \tan(\delta_0 + \varphi),$$

in which φ may be regarded as simply a constant addition to δ_0. It will be seen that, while P produces a shearing stress which varies as d^2, it develops a tensile stress varying not only as d^2 but also as $\tan(\delta_0 + \varphi)$. Since $\tan \delta_0$ increases with decreased diameter, it is evident that, with the same tensile stress in two bolts of different diameters, the shearing stress will be larger in the smaller bolt.

The disadvantage of this increased shearing stress in setting up the nuts of small bolts, is aggravated by the tendency of the average mechanic to put excessive force upon the wrench in such cases. As a result of a series of tests made at Cornell University, Professor Barr * concludes:

"(*a*) That the initial tensile load due to screwing up for a tight joint varies about as the diameter of the bolt — that is, a mechanic will graduate the pull on the wrench in

* "Notes on Machine Design," 1900, p. 106.

about that ratio. (*b*) That the load produced may be estimated at 16,000 lbs. per inch of diameter of bolt, or

$$P_1 = 16{,}000\, d,$$

in which P_1 is the initial load in pounds due to screwing up, and d is the nominal (outside) diameter of the screw thread. * * * If the initial load due to screwing up be divided by the cross-sectional area of the bolt at the bottom of the thread, the initial intensity of the tensile stress is obtained. The above experiments indicate that this intensity of stress varies, approximately, inversely as the nominal diameter (d) of the bolt; and that it may frequently equal or exceed:

$$f = \frac{30{,}000}{d} \text{ lbs. per sq. in.}$$

In addition to this tensile stress, there is a considerable twisting action on the bolt."

Mr. Harvey D. Williams * has calculated the efficiency of the U. S. Standard bolts whose proportions are given in Table XI., on the basis of the ratio between the useful fibre stress — or that portion which would be required for the support of the safe axial load only — and the total fibre stress produced in screwing up the nut. His results are given in Table XXXIII.

The method of computing the efficiencies given, is as follows:

From (55) the value of P is found in terms of W, p and d_0 being known for any given bolt and μ being taken as 0.15. Then, $P = K \cdot W$, where K is a numerical factor. Also:

$$\textit{Twisting Moment} = T.M. = P \times \frac{d_0}{2};$$

from (73):

$$S_s = \frac{16 T.M.}{\pi d^3};$$

from (60):

$$S_t = \frac{4W}{\pi d^2}.$$

Then:

$$\textit{Maximum tensile stress} = f = \tfrac{3}{8} S_t + \tfrac{5}{8}\sqrt{S_t^2 + 4S_s^2};\dagger \qquad (74)$$

$$= \frac{W}{2\pi d^2}\left[3 + 5\sqrt{1 + \left(\frac{8 T.M.}{Wd}\right)^2}\right].$$

But, to support the load, W, there is required only per sq. in. the

$$\textit{Useful tensile stress} = f' = S_t = \frac{4W}{\pi d^2}.$$

The load, W, must be reduced below the amount which the

* *Jour. Am. Soc. Naval Engineers*, Vol. XIII., No. 2.

† Lanza: "Applied Mechanics," 1897, p. 892.

screw would carry, if under direct tension only, in order that the load produced by the stress, f, shall not exceed the strength of the bolt. Hence, in this respect — *considering the thread-friction only* — the efficiency of the bolt in raising the weight W, is:

$$E = \frac{f'}{f} = 8 \div \left[3 + 5\sqrt{1 + \left(\frac{8 T.M.}{Wd} \right)^2} \right], \qquad (75)$$

in which W is the useful axial load in pounds, $T.M.$ is the torque in inch-pounds, and d is the effective diameter in inches. Equation (74) is similar to (62), the former being the formula deduced by Grashof and the latter that by Rankine.

Referring to the table, Mr. Williams says:

"The factor of safety equals the direct load factor 7, divided by the efficiency; and the safe loads given in the body of the table correspond to the factor of safety in the same horizontal line and the ultimate strength at the head of the column. To facilitate the computation of bolts having threads which are finer or coarser than the standard, the column headed "Relative Fineness of Thread" is given, in explanation of which it need only be remarked that the relative fineness of thread equals the number of threads per inch multiplied by the diameter and that bolts of different sizes but having the same relative fineness of thread will have the same efficiency and the same factor of safety. As the thread is made relatively finer and finer beyond the limits of the table, the corresponding efficiency approaches 88.06 per cent. as a limiting value, beyond which it cannot go. The factor of safety meantime approaches the limiting value, 7.95. The efficiency of a hollow bolt is always greater than that of a solid bolt of the same diameter and number of threads, the limiting efficiency for a very fine thread on a very thin tube being 96.48 per cent., as against 88.06 per cent. for a solid bolt. The error will therefore be always on the safe side, if we use the efficiencies and factors of safety given in the table in computing hollow bolts."

Seaton and Rounthwaite * give a table for the effective strength of Whitworth screws in which the torsional stress is allowed for by assuming progressively lower values for the working stress as the bolts diminish in size. The table is based on the relation:

$$\textit{Working Stress per sq. in.} = (\textit{Effective Area})^{\frac{5}{12}} \times C,$$

where $C = 5{,}000$ for iron or mild steel and 1,000 for muntz or gun-metal. For iron or steel bolts above 2 inches in diameter and gun-metal or bronze ones above 3½-inch diameter, the moment of the twisting stress is small, proportionately, and is neglected in the table, the working stresses in lbs. per sq. in., for all sizes above those noted being uniformly 7,000 and 2,500, respectively.

* "Pocketbook of Marine Engineering Rules and Tables," 1899, p. 73.

TABLE XXXIII.

SAFE LOADS FOR U. S. STANDARD BOLTS.

Relative Fineness of Thread.	Nominal Diameter of Bolt.	Number of Threads per Inch.	Efficiency. $\frac{f'}{f}\%$	Factor of Safety. $7 \div \frac{f'}{f}$	Ultimate Strength. 20,000 Alloy. Cu, 88%, Sn, 10%, Zn, 2%	40,000 Phosphor-bronze.	50,000 Wrought Iron and Best Rolled Bronze.	60,000 Class B Bolt Material.	65,000 Class A Bolt Material.	80,000 Class A, Nos. 1 and 2, Machinery Forgings.	95,000 High Grade Machinery Forgings.
5	$\frac{1}{4}$	20	74.68	9.4	57	115	143	172	186	229	272
	$\frac{5}{16}$	18	76.56	9.1	99	198	247	297	322	396	470
6	$\frac{3}{8}$	16	77.49	9	150	301	376	451	488	601	714
	$\frac{7}{16}$	14	78.38	8.9	207	415	519	623	675	830	986
6.5	$\frac{1}{2}$	13	78.48	8.9	282	564	704	845	915	1,125	1,340
	$\frac{9}{16}$	12	78.92	8.9	365	730	912	1,095	1,186	1,460	1,730
	$\frac{5}{8}$	11	79.11	8.8	456	913	1,140	1,370	1,480	1,820	2,170
7.5	$\frac{3}{4}$	10	80.00	8.8	690	1,380	1,725	2,070	2,240	2,760	3,280
	$\frac{7}{8}$	9	80.48	8.7	964	1,930	2,410	2,900	3,140	3,860	4,580
8	1	8	80.61	8.7	1,265	2,530	3,170	3,800	4,120	5,060	6,010
	$1\frac{1}{8}$	7	80.48	8.7	1,595	3,190	3,990	4,790	5,180	6,380	7,570
	$1\frac{1}{4}$	7	81.37	8.6	2,070	4,140	5,180	6,210	6,730	8,280	9,830
	$1\frac{3}{8}$	6	80.92	8.7	2,440	4,890	6,110	7,330	7,940	9,780	11,600
9	$1\frac{1}{2}$	6	81.61	8.6	3,020	6,040	7,540	9,060	9,800	12,050	14,300
	$1\frac{5}{8}$	$5\frac{1}{2}$	81.56	8.6	3,530	7,060	8,820	10,600	11,500	14,100	16,750
	$1\frac{3}{4}$	5	81.37	8.6	4,060	8,120	10,150	12,200	13,200	16,200	19,250
	$1\frac{7}{8}$	5	81.92	8.5	4,800	9,600	12,000	14,400	15,600	19,200	22,800
9	2	$4\frac{1}{2}$	81.61	8.6	5,360	10,750	13,400	16,100	17,400	21,500	25,500
	$2\frac{1}{4}$	$4\frac{1}{2}$	82.43	8.5	7,120	14,200	17,800	21,400	23,100	28,500	33,800
10	$2\frac{1}{2}$	4	82.35	8.5	8,750	17,500	21,900	26,300	28,400	35,000	41,500
11	$2\frac{3}{4}$	4	83.20	8.4	11,000	22,000	27,500	33,000	35,700	44,000	52,200
12	3	4	83.42	8.4	13,400	26,800	33,500	40,200	43,600	53,600	63,600
13	$3\frac{1}{4}$	4	83.88	8.3	16,100	32,200	40,200	48,400	52,400	64,400	76,400
14	$3\frac{1}{2}$	4	84.20	8.3	19,000	38,100	47,600	57,200	61,900	76,200	90,400
15	$3\frac{3}{4}$	4	84.47	8.3	22,200	44,500	55,600	66,700	72,300	89,000	105,500
16	4	4	84.71	8.3	25,700	51,400	64,200	77,000	83,400	102,800	122,000
17	$4\frac{1}{4}$	4	84.91	8.2	29,350	58,700	73,400	88,100	95,400	117,400	139,300
18	$4\frac{1}{2}$	4	85.09	8.2	33,300	66,600	83,200	100,000	108,000	133,000	158,000
19	$4\frac{3}{4}$	4	85.26	8.2	37,400	75,000	93,700	112,000	122,000	150,000	178,000
20	5	4	85.44	8.2	41,900	83,800	105,000	126,000	136,000	167,500	199,000
21	$5\frac{1}{4}$	4	85.55	8.2	46,600	93,200	116,500	140,000	151,000	186,000	221,000
22	$5\frac{1}{2}$	4	85.68	8.2	51,500	103,000	129,000	154,500	167,000	206,000	244,500
23	$5\frac{3}{4}$	4	85.80	8.2	56,700	113,500	142,000	170,000	184,000	227,000	269,000
24	6	4	85.92	8.1	62,000	124,000	155,000	186,000	202,000	248,000	295,000

30. Types of Screw Fastenings.

Screw fastenings have forms as numerous as their uses are varied. Brief reference will be made to a few types.

1. BOLTS, TAP BOLTS, STUDS.—The proportions of *Machine Bolts* have been given in preceding tables. When employed to

join flanges, as in Fig. 40, this form, if short and tightly fitted, gives a most rigid connection. For steam cylinder heads, they are somewhat objectionable, since, if a bolt breaks, the lagging must be removed to replace it.

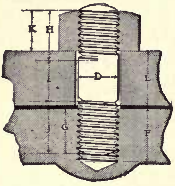


FIG. 42.

With the *Stud*, on the contrary, the broken part may be drilled out readily from the flange. The stud has further advantages in its use when through bolts are inadmissible and in the fact that, once set in the weak threads of cast metal, it need not be removed, as the tap-bolt must be, to disconnect the parts. The threaded portion which enters the casting should be longer than that for the nut and the unthreaded shank should be shorter than the flange through which it passes. Fig. 42 * gives good general proportions, as follows:

$D =$ diameter of stud;

$F = 1.25D =$ depth of hole;

$G = 1.15D =$ length of stud to be screwed in;

$H = 1.30D =$ length of thread on nut-end;

$J = F =$ length of thread on opposite end.

The *Tap-Bolt*, Fig. 43, is practically a machine-bolt without a nut, the shank passing through a flange or other member and the threaded section screwing into the remaining part connected. Like the stud, it is liable to stick fast and it has the further disadvantage that its frequent removal to break the joint will wear the weak threads of a casting. For this reason, the depth of the tapped hole should be from 1.5 to twice the diameter. The proportions of counter-sunk and round and button-head tap-bolts and screws are given in Table XXXIV.


FIG. 43.

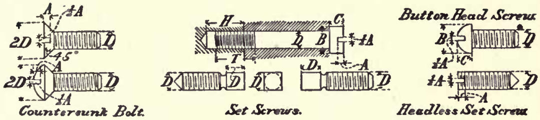
2. SET-SCREWS are fastenings which are suitable only for light work. They find most frequent use in securing pulleys, etc., to shafting. Their chief advantage is that no key-way is necessary and that, therefore, the connected piece may be readily shifted. The disadvantages are the liability to slip, the burring of the shaft,

* *American Machinist*, June 6, 1901.

Table XXXIV.

Tap-Bolts and Set-Screws.

(Newport News Shipbuilding and Dry Dock Co.)



Diameter, D	Threads, per Inch.	Tap Bolts.		Round Heads.		Button Heads.		Slot.	
		Tap Hole, H	Length of Thread, T	Diameter, B	Depth of Head, C.	Diameter, B	Depth of Head, C	Depth, A	Width, $\frac{1}{2}A$
$\frac{1}{4}$	20	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{3}{64}$
$\frac{5}{16}$	18	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{15}{32}$	$\frac{15}{64}$	$\frac{15}{32}$	$\frac{15}{64}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{3}{8}$	16	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{9}{32}$	$\frac{9}{16}$	$\frac{9}{32}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{7}{16}$	14	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{21}{64}$	$\frac{5}{8}$	$\frac{21}{64}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{1}{2}$	13	1	1	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{9}{16}$	12	$1\frac{1}{16}$	$1\frac{1}{16}$	$\frac{27}{32}$	$\frac{27}{64}$	$\frac{27}{32}$	$\frac{27}{64}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{5}{8}$	11	$1\frac{3}{16}$	$1\frac{3}{16}$	$\frac{15}{16}$	$\frac{15}{32}$	$\frac{15}{16}$	$\frac{15}{32}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{3}{4}$	10	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{16}$	$\frac{9}{16}$	$1\frac{1}{16}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{1}{8}$
$\frac{7}{8}$	9	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{1}{4}$	$\frac{21}{32}$			$\frac{1}{4}$	$\frac{1}{8}$
1	8	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{7}{16}$	$\frac{3}{4}$			$\frac{5}{16}$	$\frac{5}{32}$
$1\frac{1}{8}$	7	$1\frac{15}{16}$	$1\frac{15}{16}$	$1\frac{5}{8}$	$\frac{27}{32}$			$\frac{5}{16}$	$\frac{5}{32}$
$1\frac{1}{4}$	7	$2\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{13}{16}$	$\frac{15}{16}$			$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{3}{8}$	6	$2\frac{5}{16}$	$2\frac{5}{16}$	2	$1\frac{1}{32}$			$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{1}{2}$	6	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{3}{16}$	$1\frac{1}{8}$			$\frac{7}{16}$	$\frac{7}{32}$
$1\frac{5}{8}$	$5\frac{1}{2}$	$2\frac{11}{16}$	$2\frac{11}{16}$						
$1\frac{3}{4}$	5	$2\frac{7}{8}$	$2\frac{7}{8}$						
$1\frac{7}{8}$	5	$3\frac{1}{16}$	$3\frac{1}{16}$						
2	$4\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{4}$						
$2\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{8}$						
$2\frac{1}{2}$	4	4	4						
$2\frac{3}{4}$	4	$4\frac{3}{8}$	$4\frac{3}{8}$						
3	4	$4\frac{3}{4}$	$4\frac{3}{4}$						

the radial stress in the hub, and the uneven bearing and slight eccentricity of the latter, if a free fit. The points of set-screws are made flat, conical, rounded, or cupped. A shallow hole is sometimes bored in the shaft to receive the point. In light work, however, the screw is set up sufficiently to make its own indentation. A relatively strong fastening may be made by interposing a thin steel plate between the set-screw and a "flat" filed on the shaft, the plate fitting into a recess in the hub.

Professor Lanza * tested the holding power of points of various forms upon a $\frac{11}{16}$-in. shaft, the screws being of wrought iron, $\frac{5}{8}$-in. diameter, 10 threads to the inch, and set up with a force of 75

* *Trans. Am. Soc. Mech. Engs.*, Vol. X.

lbs. at the end of a 10-in. monkey wrench. The shaft was of steel and the points made but little impression upon it. Two screws were used to secure a pulley to the shaft and then the circumferential load required to make the pulley slip was found, from which load the resistance of the screws was determined. The shapes of the points were:

A. Ends perfectly flat, $\frac{9}{16}$ in. diameter.
B. Ends rounded, radius $\frac{1}{2}$ in.
C. Ends rounded, radius $\frac{1}{4}$ in.
D. Ends cup-shaped and case-hardened.

The holding power in pounds was:

	Lowest.	Highest.	Average.
A.	1412	2294	2064
B.	2747	3079	2912
C.	1902	3079	2573
D.	1962	2958	2470

Professor Lanza states as to:

A. The set-screws were not entirely normal to the shaft; hence they bore less in the earlier trials before they had become flattened by wear.

B. The ends of these set-screws, after the first two trials, were found to be flattened, the flattened area having a diameter of about $\frac{1}{4}$ in.

C. The ends were found, after the first two trials, to be flattened, as in *B*.

D. The first test held well because the edges were sharp; then the holding power fell off till they had become flattened in a manner similar to *B*, when the holding power increased again.

3. Eye-Bolts. — Good proportions for eye-bolts are given in Table XXXV. Since the bolt when screwed home is without load, the torsional effect is negligible and the same working stress may be used for all sizes. Owing to bending action, the sides of the eye are subjected to greater stress than the body of the bolt, and their combined cross-sectional area is made greater than that of the weakest section at *G*, the excess being about 200 per cent. in the $\frac{3}{8}$-inch bolt and decreasing rapidly with the larger sizes.

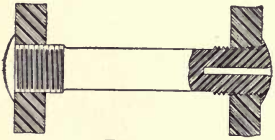


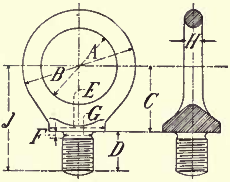
Fig. 44.

4. Stay-Bolts.—These bolts are used to brace the flat surfaces of boilers. They vary in details of form and manufacture. Good practice is shown by Fig. 44. The bolt is threaded at each end, turned down in the shank to the diameter at the base of the thread, screwed into both sheets, and riveted over cold with shallow spherical heads. Minimum general proportions are: diameter,

TABLE XXXV.

EYE-BOLTS.

(UNION IRON WORKS.)



Size.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	Capacity Based on 10,000 lbs. per sq. Inch Strain.
3/8	1 1/2	2 1/4	1 3/8	5/8	1	3/32	5/16	3/8	2	767
1/2	1 1/2	2 1/4	1 3/8	5/8	1	3/32	3/8	3/8	2	1,104
5/8	2	3	1 3/4	1	1 1/4	1/8	1/2	1/2	2 3/4	1,963
3/4	2	3	1 3/4	1	1 1/4	1/8	9/16	1/2	2 3/4	2,485
7/8	2 1/4	3 1/2	2 1/8	1 1/4	1 3/4	1/8	11/16	5/8	3 3/8	3,712
1	2 1/4	3 1/2	2 1/8	1 1/4	1 3/4	1/8	13/16	5/8	3 3/8	5,185
1 1/8	2 1/2	4	2 1/2	1 1/2	2 1/4	8/16	15/16	3/4	4	6,903
1 1/4	2 1/2	4	2 1/2	1 1/2	2 1/4	3/16	1	3/4	4	7,854
1 3/8	2 3/4	4 1/2	2 3/4	2	2 1/2	3/16	1 1/16	7/8	4 3/4	9,940
1 1/2	2 3/4	4 1/2	2 3/4	2	2 1/2	3/16	1 1/4	7/8	4 3/4	12,270
1 5/8	3 1/2	5 3/4	3 1/2	2 1/2	3	1/4	1 5/16	1 1/8	6	13,520
1 3/4	3 1/2	5 3/4	3 1/2	2 1/2	3	1/4	1 7/16	1 1/8	6	16,210
1 7/8	4	6 1/2	4	3	3 1/2	1/4	1 9/16	1 1/4	7	19,150
2	4	6 1/2	4	3	3 1/2	1/4	1 11/16	1 1/4	7	22,340

$\frac{7}{8}$ inch; threads per inch, 12; spacing, centre to centre, 4 inches. The stress at root of thread should not exceed 6,000 lbs. per sq. in. A "detector" hole—at which leakage will show when the bolt is broken—is drilled or punched, preferably the former, from the outer end of the bolt inward to the beginning of the shank.

Flexibility is a most important requirement of these bolts. In some cases, various combinations of the ball-and-socket joint have been applied at one end. In the ordinary type, this quality depends upon the material, the reduced shank, and the form and method of driving the heads. As material, the best grade of wrought iron is preferred.

The Falls Hollow Staybolt is rolled with a central hole throughout, thus avoiding later drilling or punching. The bolt is also threaded through its full length with, therefore, uniform strength at all points. The size of the hole is usually $\frac{1}{8}$ inch or $\frac{3}{16}$ inch. It serves not only as a "detector" but also, if desired, as an inlet for the admission of air to aid combustion.

The data and results of tests of these bolts at McGill University are:

Material, double-refined charcoal stay-bolt iron, 1 inch diameter, $\frac{3}{16}$ inch hole; length, $25\frac{3}{8}$ inch; mean diameter, outside, 1.014 inch; yield-point, 32,000 lbs. per sq. in.; ultimate tensile strength, 49,300 lbs. per sq. in.; equivalent elongation in 8 inches, $31\frac{3}{16}$ per cent.; reduction of area, 45.7 per cent.

Chief Engineers Sprague and Tower, U. S. Navy, in 1879, made exhaustive experiments upon the strength of boiler-bracing. From their report * the following data are taken with regard to the resistance of screw stay-bolts in flat surfaces:

"In reference to iron and low steel bolts, and iron and low steel plates, and copper plates and iron bolts, after a careful examination of the results of these experiments in particular, we are satisfied that the following formulæ will correctly and safely represent the working strength of good material in flat surfaces, supported by screw stay-bolts with riveted button-shaped heads or with nuts, when the thickness of the plates forming said surfaces and the screw stay-bolts are made in accordance with the dimensions and conditions given in Table Y. W = safe-working pressure; T = thickness of plate; d = distance from centre to centre of stay-bolt:

For iron plates and iron bolts. . . . . $W = 24000 \dfrac{T^2}{d^2}$

For low steel plates and iron bolts . . . . $W = 25000 \dfrac{T^2}{d^2}$

For low steel plates and low steel bolts. . . . . $W = 28000 \dfrac{T^2}{d^2}$

For iron plates and iron bolts, with nuts . . . . $W = 40000 \dfrac{T^2}{d^2}$

For copper plates and iron bolts. . . . . $W = 14500 \dfrac{T^2}{d^2}$

"To obtain the ultimate bursting pressure, multiply the results of the above formulæ by 8, which is the factor of safety used.

TABLE Y.

DIMENSIONS AND CONDITIONS FOR MAKING IRON AND LOW STEEL SCREW STAY-BOLTS FOR FLAT SURFACES SUBJECT TO INTERNAL PRESSURE FOR DISTANCES RANGING FROM FOUR TO EIGHT INCHES (INCLUSIVE) FROM CENTRE TO CENTRE OF STAY-BOLT.

Thickness of Plate.	Diameter of Bolt Outside of Thread.	Number of Threads Per Inch.	Length of Bolt Left Through for Riveting in Fractions of Chin of Bolt.	Height of Rivet-head when Finished.	Diameter of Base of Rivet-head Not to Exceed when Finished.	Nuts.	
						Breadth of Annular Bearing Surface.	Dished Out to a Depth of—
$\frac{1}{4}''$	$1''$	14	$\frac{1}{2}''$	$\frac{7}{16}''$	$1\frac{5}{16}''$	$\frac{1}{16}''$	$\frac{1}{16}''$
$\frac{3}{8}$	$1\frac{1}{8}$	14	$\frac{1}{2}$	$\frac{1}{2}$	$1\frac{9}{16}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{1}{2}$	$1\frac{1}{4}$	12	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{3}{4}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{5}{8}$	$1\frac{3}{8}$	12	$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{7}{8}$	$\frac{1}{4}$	$\frac{3}{32}$

* "Experiments in Boiler Bracing," U. S. Navy Dep't, 1879.

"The rivet-heads to be a segment of a sphere, formed by first upsetting the end of the bolt with a few quick, sharp blows of the hammer, then finished to shape with the hammer and button-head set. Where nuts *can* be used instead of riveted heads, they should be of the standard size, suited to the diameter of the bolt, faced on the side bearing on the plate, and dished out so as to form an annular bearing surface of as large a diameter as the nut will allow, aud of a breadth and depth given in the table. Before securing the nut in place the dished portion should be filled with red-lead putty made stiff with *fine* iron borings."

The regulations (January, 1901) of the U. S. Board of Supervising Inspectors of Steam Vessels, prescribe for plates, $\frac{7}{16}$ inch thick and under, used in boilers as "flat surfaces fitted with screw stay-bolts riveted over, screw stay-bolts and nuts, or plain bolt with single nut and socket, or riveted head and socket," a working pressure determined by the formula:

$$P = \frac{C \times t^2}{d^2}, \tag{76}$$

where $P =$ working pressure in lbs., $C = 112$, $t =$ number of sixteenths in plate thickness (*i. e.*, for $\frac{7}{16}$-inch plate, $t = 7$), and $d =$ distance between stays in inches. For plates above $\frac{7}{16}$-inch thick $C = 120$. The pressures, as above, refer to fire-box plates. Also,

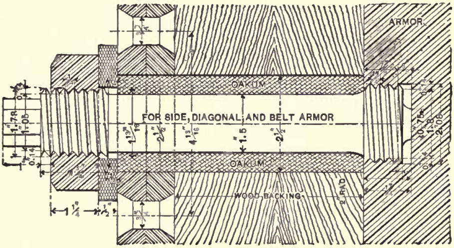


FIG. 45.

"on other flat surfaces there may be used stay-bolts with ends threaded, having nuts on same, both on the outside and inside of plates." For these surfaces, formula (76) is used with $C = 140$.

5. ARMOR BOLTS.—The proportions of threads for these bolts, as used in the U. S. Navy, have been given in § 24. The method of their application with side, diagonal, and belt-armor, is illustrated in Fig. 45. The armor-plate is fitted snugly to a backing of teak, the latter being secured to the backing plates of the hull by bolts countersunk in the wood. After the armor-bolt is screwed down

to a bearing in the plate, the space around the shank is calked solidly with oakum and the nut is screwed up against a lead washer until it embeds itself in the latter, thus causing the lead to flow into the thread. As an additional precaution against leakage, the backing plates and washer are coated with red lead, all interstices in the backing are filled with red lead under pressure, and the joint between the backing plates is calked. Turret-armor is secured by similar bolts which have, however, a solid head instead of a nut. The spacing, in all cases, is such as to provide one bolt for each 5 sq. ft. of armor surface.

31. Methods of Manufacture.

Bolts are headed hot from round stock; then threaded and pointed. Nuts are pressed or forged hot, or pressed and punched cold, and tapped.

1. Bolt-Blanks. — The round stock is sheared into lengths containing enough material for shank and head. Each blank is then heated and the head formed in a forging machine. Figs. 46 and 46*a* give a view, plan, and details of the 1½-inch Heading and Forging Machine built by the Acme Machinery Co., Cleveland, O., and illustrated herein through the courtesy of that company.

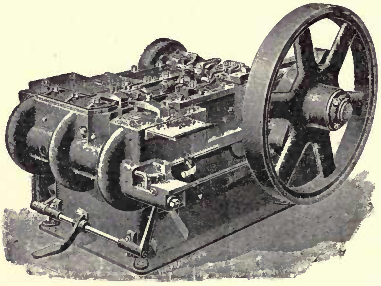


Fig. 46.

"There are two sets of tools: the stationary or gripping dies, *A*, which hold and release the blank and the heading die, *B*, and finishing punch, *C*, which form the head and are carried by a tool-holder fixed to a reciprocating plunger. The latter is driven from the shaft which is actuated by a fly-wheel with clutch-connection controlled by a pedal. The plunging or upsetting mechanism is omitted from the plan; it moves on the line marked "centre of heading slide."

"The dies, *A*, are divided and open vertically on the centre-line of the lower cylindrical groove, *D*, and the upper groove, *E*, also cylindrical but having a square or hexagonal recess for the bolt-head. The opening and closing of the dies is done by the toggle-joint mechanism shown. The latter is operated, through an intervening spring, by an adjustable connecting rod driven from the shaft. The bolt-blank is upset while in groove, *D*, by the heading die, *B*. It is then shifted to *E* where the head is finished by punch, *C*. The grooves, *D* and *E*, are concentric respectively with die, *B*, and

punch, *C.* The latter die holds a die-plug and the punch has a head, both suitably shaped for upsetting square, or with other forms, hexagonal heads.

"In forging a bolt-head, the operator places a heated blank in groove, *D*, and touches the pedal. The machine makes a "plunge" and the gripping dies close, remaining thus while die, *B*, advances and forms the head and until the plunger has travelled about 3 inches. When the machine has passed its forward centre, the plunger has receded about ¾ inch and the gripping dies open. The operator now removes the bolt to the upper groove, *E*, and again touches the pedal, upon which the finishing punch enters the die at *E*, presses against the head, and removes the slight draught formed during the first stroke. At the same time, the side pressure of the dies drives all 'fins' back into the head. The bolt is really made during the first stroke, while the heated metal is at its best for working. The second stroke simply removes the slight taper of the head and smooths the sides of the latter.

"The toggle-joint gives maximum pressure when the gripping dies are closed. Until its joints are in line, it is acted upon by an elastic force in the spring, so that if the dies become obstructed, the mechanism will yield and the machine will not meet undue

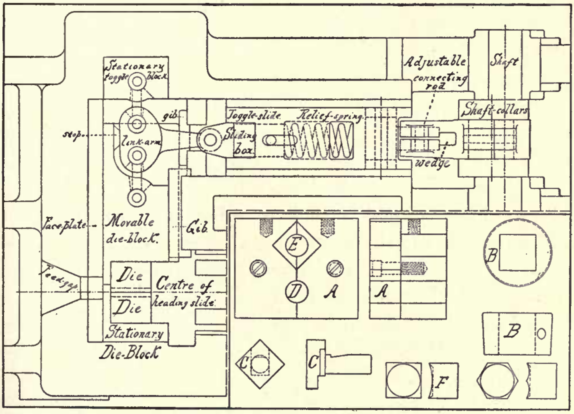


FIG. 46*a*.

strain. In addition to its action as an automatic relief, the spring forms also, with the connecting rod of adjustable length, a device to regulate the time and duration of closure of the gripping dies with regard to the advance of the heading dies on the plunger, as may be required for various sizes of work."

The machine described above is of the "grip-and-plunge" type. In the "hammer-header" form of heading machine, there are, for a square head, five hammers, one striking on the top and one on each of the four faces of the head, simultaneously. In this machine, the head is molded by a succession of relatively light blows while it is cooling. An objection urged against this form is that

the bond between the head and shank may be destroyed by a "cold-shut" at the point of juncture.

2. Nut-Blanks are made by several processes. In the "hot pressed" machine, the nut is formed in a die, pierced, and crowned, and is then placed in the holder of a "burring machine" in which revolving cutters remove the rough edges. In "hot forging machines," the nut is forged smooth by hammers automatically operated; and, in "cold pressing," the flat bar is fed between the rolls of the machine, cut into blanks, and a nut made complete at each revolution. Finally, if desired, the nut is faced and chamfered in a facing machine.

While the cold-punched nut meets extensive service in structural and other work, the rigid specifications of the Bureau of Steam Engineering, U. S. Navy, permit the use of hot-pressed nuts only. With regard to this question, the Engineer-in-chief says:

"In making a cold-punched nut, either of wrought iron or steel, the fibre of the metal is injured and its full strength can be restored only by bringing the nut to a welding heat and finishing it under the hammer, as with the hand-made forged nut. The hot-pressed nut, on the contrary, although not so perfect as that made by hand forging, approaches the latter so nearly that it can be reamed, tapped, finished, and used with fair degree of safety."

The injurious effects upon boiler-plate of punching rivet-holes will be discussed in the succeeding chapter. In 1878, Mr. David Townsend * made some experiments which show the flow of metal in nuts punched cold under the conditions of his test. He found that both the top (nearest the punch) and bottom faces of the nut were depressed; that the lower diameter was increased, making the sides tapering; and that a portion of the blank punched from the hole had flowed into the body of the nut throughout a zone nearly half as deep as the nut and beginning almost at the top face of the latter. The original depth of the nut was 1.75 in.; that of the core removed was 1.063 in. The density of the latter was found to be the same as that of the metal before punching. Therefore, a volume of metal, whose sectional area was that of the core and whose length was $1.75 - 1.063 = 0.687$ ins. was forced into the body of the nut. It is apparent that the stress was severe.

3. Threading and Tapping. — Bolts are threaded in the lathe, or by hand-operated dies, or in the bolt-cutter, the latter being

* *Jour. Franklin Institute*, March, 1878.

practically but a set of revolving dies into which the bolt-blank is fed at the required axial speed. The bolt-cutter produces usually a full thread at one cut with, in consequence, greater stress in the bolt metal and greater pressure upon the lead-screw than in the lathe where the same thread would be made in several cuts. Square threads or those requiring unusual accuracy of workmanship require lathe-work. The merits of the bolt-cutter lie in the rapidity and cheapness of execution and the fact that its product is sufficiently accurate for all ordinary purposes.

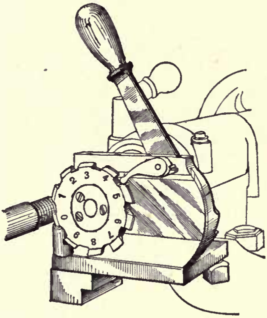


FIG. 47.

Fig. 47 shows a threading tool which is illustrated herein through the courtesy of the Rivet-Dock Company, Boston, Mass. The disadvantages of the single-point thread tool used in lathe work are: the difficulties of keeping the exact angle in grinding, of setting with the small thread-gauge, the succession of cuts, necessarily light, to prevent burning the point, and the repeated stops to test with a limit-gauge or master-nut.

The thread-cutter shown, is a simple disc of tool steel having ten teeth, each of the latter being longer radially than the one preceding. In operation, a cut is run with each tooth. There are thus, in effect, ten cutting tools, the leading ones suitably shaped for roughing out and the final tooth proportioned for finishing with accuracy. The single-point tool both roughs and finishes, while the final tooth of the cutter does finishing work only. The cutter is mounted on a steel slide, the latter having a movement to and from the work by means of an eccentric stud in the hub of the lever. The lever, in moving the slide, engages the pawl and rotates the cutter one tooth for the next cut. The heel of the tooth in action rests upon a stop, which takes the strain of cutting. The stud extends through the lever-hub and is secured on the back by an arm with pin-stop engaging ten holes so spaced that changing the stop from one hole to another moves the slide and cutter a fraction of a thousandth of an inch forward, thus giving the necessary adjustment for fine fits and provision for exact duplication.

Bolt-threads are produced also by cold rolling. For the description of this process which follows, acknowledgment is due to J. H. Sternbergh, Esq., President of the American Iron and Steel Manufacturing Company.

"The machine is horizontal and of simple construction. It has a stationary die with threads cut on the face of the latter at a certain angle. Another die, having threads cut on its face also, is held in a reciprocating cross-head. The bolt-blank is placed perpendicularly between the two dies and the thread is produced by compression in rolling the blank between the latter. The distance between the apices of the dies is the same as the diameter of the bolt at the root of the thread. For a bolt of, say, $\frac{3}{4}$-inch diameter, the dies are about ten inches long and the blank is rolled throughout nearly the whole length of the die, one operation producing the thread. A portion of the latter is actually raised above the external circumference of the bolt and no metal whatever is cut away. Great accuracy, however, is required as to the diameter of the blank bolt in order to produce uniform and perfect threads."

There are various types of machines for threading nuts. In one well-known automatic nut-tapper, the blank nuts are placed in a receptacle on the top, from which they are conveyed to the taps by means of guide-ways. After being threaded, the nuts are ejected automatically. It is stated that one operator can attend ten machines and produce about 180,000 nuts per day.

32. Materials.

The specifications (1901) of the Bureau of Steam Engineering, U. S. Navy, for bolts and nuts of steel and iron are as follows:

RODS FOR BOLTS, STUDS, AND RIVETS.

1. The physical and chemical characteristics of rods for bolts, studs, and rivets are to be in accordance with the following table:

Class.	Material.	Minimum Tensile Strength.	Minimum Elastic Limit.	Minimum Elongation.	Maximum Amount of —		
					P.	S.	
		Lbs. per sq. in.	*Lbs. per sq. in.*	*Per cent. in 8 Inches.*			
Class A.	Open-hearth nickel or carbon steel.	75,000	40,000	23	.04	.03	Cold and quench bend about an inner diameter equal to the thickness of the test piece in each case. Quenching temperature 80° to 90° F.
Class B.	Open-hearth carbon steel.	58,000	30,000	28	.04	.03	Inner diameter equal to one half the thickness.

If the contractor desires, and so states on his orders, the Bureau will direct that the inspection of the rods be made at the place of manufacture of the bolts, studs or rivets instead of at the place where the rods are rolled.

2. *Kind of Material.* — The steel shall be made by the open-hearth process, shall contain not more than four one-hundredths of 1 per cent. of phosphorus, nor more than three one-hundredths of 1 per cent. of sulphur.

3. *Surface and other Defects.* — The rods must be true to form, free from seams, hard spots, brittleness, injurious sand or scale marks, and injurious defects generally.

4. *Test Pieces.* — If the total weight of rods, all of the same diameter, and rolled from the same heat, amounts to more than 6 tons, the inspector shall select at random six tensile test pieces, three cold-bending test pieces and three quench-bending pieces; but if the weight is less than 6 tons, one half of that number of test-pieces will suffice. If, however, the rods in one heat are not of the same diameter, then the inspector will take such additional test pieces as he may consider necessary according to the number of different sizes of rods in the heat. All of the test pieces shall be taken from rods finished in the rolls and, when practicable, but one piece will be cut from each rod selected for test. Should any test piece be found too large in diameter for the testing machine, the piece may be prepared for test in the manner prescribed for forgings. The tensile tests for rounds ⅝ inch in diameter and less, shall be made on the largest sizes available and the elongation measured on a length equal to eight times the diameter.

5. *Bending Tests.* — The cold and quench test pieces of Class A1 rods shall stand bending through an angle of 180° around a curve, the inner diameter of which is equal to the diameter of the rod. The cold and quench bends of Class A2 rods shall stand bending through an angle of 180° around a curve, the inner diameter of which is equal to one half the diameter of the rod. The quench test piece shall be heated to a dark cherry red in daylight, and plunged into fresh clean water at a temperature between 80° and 90° F. No bending test will be satisfactory if any cracks are to be seen on the outside of the bent portion.

Finished Bolts, Studs, and Rivets, Classes A and B.

After the rods to be made up into bolts, studs, and rivets have been tested as previously described, the finished articles shall be tested by lots of 500 pounds or fraction thereof, one piece being taken to represent the lot. The failure of 10 per cent. of the lots of 500 pounds to stand the specified tests in a satisfactory manner will render the whole of any delivery liable to rejection.

Bolts and Studs. — When the bolts or studs are of sufficient length in the plain part to admit of being bent cold, they must stand bending double to a curve of which the inner radius is equal to the radius of the bolt or stud, without fracture.

When bolts or studs are not of sufficient length in the plain part to admit of being bent cold, the threaded part must stand bending cold without fracture as follows:

If of ½ inch diameter or less	35°
If above ½ inch diameter and under 1 inch.	30°
If 1 inch diameter or over.	25°

Where the bending tests can not be applied, the two following hammer tests must be substituted:

(*a*) The test piece to stand flattening out cold to a thickness equal to one half its original diameter without showing cracks.

(*b*) The test piece to stand flattening out, while heated to a cherry-red heat, to a thickness equal to one third its original diameter without showing cracks.

Surface Inspection. — (1) All bolts and studs shall be free from surface defects.

(2) All bolts are to be headed hot, and the heads made in accordance with the U. S. standard proportions unless otherwise specified. The head must be concentric with the body of the bolt.

(3) The threads must be of the U. S. standard unless otherwise specified, and must be clean and sharp. The threads of Classes A and B bolts may be either chased or cut with a die, but the threads of body-bound bolts must be chased and must extend far enough down so that when the nut is screwed home there will be not more than one and one half threads under it. The plain part of body-bound bolts must be turned in a lathe to fit accurately in the bolt hole.

Steel and Iron Nuts.

(*To be used with class A and B bolts and studs.*)

1. One tensile and one bending test bar from each lot of 1,000 pounds of material or less from which nuts are to be made shall be selected by the inspector for test.

2. The material (whether steel or iron) shall show a tensile strength of at least 48,000 pounds per square inch and an elongation of at least 25 per cent. in 8 inches. A bar ½ inch square or ½ inch in diameter shall bend back cold through an angle of 180° without showing signs of fracture.

3. The nuts must be free from surface defects and the threads clean, sharp, and well fitting.

4. The dimensions of threads must be in conformity with the United States standard unless otherwise specified.

5. The nuts must be hot-pressed and reamed before threading, the holes to be central and square with the faces. All nuts must fit on the bolts without shake.

Forgings.

1. The physical and chemical characteristics are to be in accordance with the following table :

Class.	Material.	Treatment.	Minimum Tensile Strength.	Minimum Elastic Limit.	Minimum Elongation.	Maximum Amount of—		Cold Bend About an Inner Diameter of—
						P.	S.	
			Lbs. per sq. in.	*Lbs. per sq. in.*	*Per cent. in 2 in.*			
High Grade,	Open-hearth nickel steel.	Annealed and oil tempered.	95,000	65,000	21	.06	.04	One inch through 180°.
Class A.	Open-hearth either nickel or carbon steel.	Annealed. Oil tempered optional.	80,000	50,000	25	.06	.04	One inch through 180°.
Class B.	Open-hearth carbon steel.	Annealed.	60,000	30,000	30	.06	.04	Half inch through 180°.

6. Nuts to be used about machinery must fit so tight that it will be necessary to use a wrench to turn them. All other nuts must be at least thumb tight.

7. For the purpose of test all nuts which fulfill the preceding requirements will be divided into lots of 500 pounds or less, and two nuts from each lot selected by the inspector for test as follows :

(*a*) One of the two shall stand flattening out cold to a thickness equal to one half its original thickness without showing cracks.

(*b*) The other shall stand flattening out (when heated to a cherry red in daylight), to a thickness equal to one third of its original thickness without showing cracks.

8. The failure to stand these tests will subject the lot represented by them to rejection. The failure of 10 per cent. of the lot to pass the tests will render the whole order liable to rejection.

For bolts requiring unusual strength, the metals described under "Forgings" are specified. Thus, connecting rod bolts are made from "High Grade" forgings as above.

For wrought iron and various alloys and bronzes, the maximum tensile strength per sq. in. of cross section is taken as:

Wrought Iron	50,000 lbs.
Alloy: Cu 88%, Sn 10%, Zn 2%	20,000 "
Phosphor Bronze, rolled	40,000 "
Muntz Metal, rolled	40,000 "
Manganese Bronze, rolled	50,000 "
Tobin Bronze	50,000 "
Naval Brass	50,000 "

The specifications (1900), of a prominent railroad company fix requirements for stay-bolt iron, as follows:

"The material desired is fagoted iron, free from admixture of steel and preferably box piled, the filling of the box being small rods. It shall show when nicked on either side and then broken, a fracture with long fiber with sound welds. The iron must be smoothly rolled, free from slivers and depressions, and shall be truly round within .01 of one inch. It shall not be more than .005 of one inch above and not more than .010 of an inch below nominal size. This to insure freedom from jamming in the threading dies.

"Sample bars will be required to meet the following physical test: They shall show when tested in full size as rolled, a tensile strength of not less than 48,000 pounds per square inch, with an elongation of not less than 25 per cent. in 8 inches. One piece from each of the two sample bars shall be subjected to tensile test and one piece from each of them shall be threaded in dies with a sharp "V" thread 12 to one inch and firmly screwed through two holders, having a clear space between them of 5 inches. One of the holders shall be of such form and length that the bolt shall be rigidly held, so as to prevent rocking. This holder will be rigidly secured to the bed of a suitable machine and the holder at the other end will be vibrated in a direction at right angles to the axis over a space of ¼ of an inch, so that the end of the specimen shall be deflected alternately ⅛ of an inch on each side of the center line. When thus tested acceptable iron should show not less than 2,200 double vibrations before breakage.

"If the test of either of the bars shows a tensile strength of less than 48,000 pounds per square inch in an original section or an elongation less than 25 per cent. in a section originally 8 inches long, or if either bar stands less than 1,700 double vibrations, or if the two give an average of less than 1,900 double vibrations before breakage, the pile represented by such two bars will be rejected and returned to the maker. In addition, those bars which fail to meet the requirements as to rolling will also be rejected and returned."

33. Nut-Locks.

As has been shown previously, the pitch-angle of screw-fastenings is so small that the screw cannot possibly "overhaul," *i. e.*, no *static* axial load, however great, will cause the nut to back off. On the other hand, on such a screw, when exposed to *shock* or to repeated, even though small, *vibrations*, the nut will loosen inevitably. Dr. Weisbach * has discussed fully and clearly the effect upon the nut of these external forces. When the joint is subjected to shock or vibration, work is done upon all of its parts. The work transferred to the nut, expends itself in producing elastic oscillations in the material of the latter, with corresponding stresses of tension or compression, and, therefore, at any instant, a resultant stress. When the moment of this resultant is equal to the moment of nut-friction, any further shock or vibration will cause the nut to yield. It is evident, therefore, that the force with which the nut is screwed home, fixes the magnitude of the shock which will loosen it. Hence, nuts which can be set up with but moderate pressure, as on shaft bearings, especially need locking arrangements. Again, the effect of small vibrations, if they follow each other with sufficient frequency, seems to be cumulative, so that, even when nuts are set up with the greatest permissible force on solid supports, as the fish-plates of rails, they will, if unlocked, back off under these conditions. The usual devices for locking a fastening nut are the check-nut, set-screws, spring-washers, and lock-plates. The nut itself is sometimes made elastic or the thread self-locking.

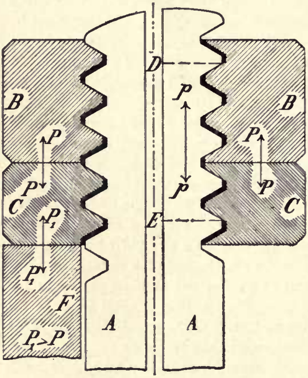


FIG. 48.

1. CHECK-NUTS.—A check-nut is essentially a friction-brake for the fastening nut. Assume, as in Fig. 48, a bolt, *A*, with fastening and check nuts, *B* and *C*, respectively. Let the lower nut be held and the upper be screwed against it as tightly as

* "Mechanics of Engineering," Vol. III., Part I., Sec. II., 1896, p. 605.

the strength of the bolt permits. There will be developed a pressure, P, between the adjoining nut-faces, and the nuts B and C will transmit this pressure to the lower and upper surfaces, respectively, of the bolt threads. Hence, a unit tensile stress, p, will exist within the bolt section, D-E, included between the limits of action of the nuts upon the bolt. This stress will not be present elsewhere. If now the bolt be subjected to shock, the fastening nut, B, cannot back off unless either its own thread-friction and that of the lower nut be overcome and the two withdraw together, or the nut, B, have a sufficient impulse to move independently, despite the friction between the nut-faces. In either case, the check-nut acts as a brake upon the fastening nut.

Again, assume, as in the left-hand half of the figure, that the bolt is used for securing the cap, F, of a shaft-bearing. Let the lower nut, C, be first screwed down until the required pressure, P_1, is produced upon the cap and then the upper nut, B, be screwed home as before, developing the pressure, P, between the nut-faces. The nut, C, is now subjected to a downward force, P, and an upward force, P_1, with a resultant, $P - P_1$, acting on the bolt-threads. There are three possible cases:

(*a*) If $P_1 > P$, the resultant force is upward, the lower nut bears on the lower surfaces of the bolt-threads and aids in sustaining the axial load, P_1, upon the bolt.

(*b*) If $P_1 = P$, the resultant force is zero. Hence the lower nut is unloaded and has no pressure on either the upper or lower surfaces of the bolt threads.

(*c*) If $P_1 < P$, the resultant force is downward, the lower nut bears on the upper surfaces of the bolt-threads, does not aid in sustaining the axial load, and produces an additional tensile stress, as at D-E.

In both (*a*) and (*c*), thread-friction exists with the lower nut and the latter acts as a brake; in (*a*) only this nut aids in sustaining the axial load, P_1, upon the bolt, which load in (*b*) and (*c*) is borne wholly by the upper nut. The fastening and check-nuts are frequently of different thicknesses. The discussion as above — adapted largely from Weisbach — shows that the upper nut may bear the entire load and should be the thicker of the two, although, in the absence of a thin wrench, the reverse is often the case. In practice, the lower nut is screwed down and home and the upper nut almost entirely so. Then, the latter is held with the wrench

and the nut, *C*, is forced backward through a slight angle until it binds on the upper nut.

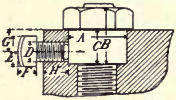


FIG. 49.

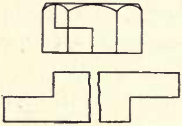


FIG. 50.

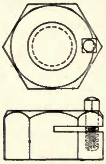


FIG. 51.

2. SET-SCREWS. — The set-screw, bearing upon a cylindrical prolongation of the nut, is the most effective locking device for heavy nuts requiring to be frequently removed as, for example, those on the connecting rods and main bearing caps of marine engines. Fig. 35 shows a bolt for the former which is fitted with a "collar-nut" and two set-screws — one for locking the nut, the other for holding the bolt when backing off the nut. Table XXXVI. gives the proportions of such collar nuts and of the dowelled stop-ring into which the set-screw is tapped. Fig. 49 shows a similar nut, omitting the stop-ring and groove, the proportions for typical sizes of which, in inches, are:

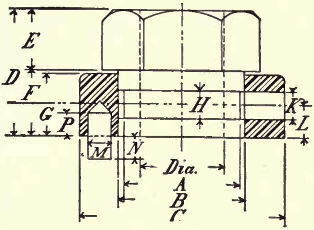
Diameter of Bolt.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	Least Value of *H* for	
								Wrought Iron or Brass.	Cast Iron.
$\frac{1}{2}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
2	$3\frac{1}{16}$	$2\frac{1}{2}$	1	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$
4	6	4	$1\frac{1}{2}$	1	$1\frac{1}{4}$	$\frac{13}{16}$	1	1	$1\frac{1}{4}$
6	9	6	2	$1\frac{3}{8}$	$1\frac{5}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{5}{8}$

3. ELASTIC NUTS. — Fig. 50 shows the nut made by the National Elastic Nut Company. The blank is cut from a flat steel bar, bent into a ring with a lap on the side, pressed in a die into the shape of a finished nut, and finally tapped with special minus taps, $\frac{1}{128}$ under size. When screwed on the bolt, the split side is forced open about $\frac{1}{128}$ of an inch, giving the nut a constant grip.

The Wiles lock-nut, shown in Fig. 51, has a slot milled half way through of a width equal to the pitch. When the nut is in place, the walls of the slot are brought slightly together by a set-screw, thus gripping the bolt-thread.

Table XXXVI.

Collar-Nuts with Locking Screws.
(Union Iron Works.)



Diam. of Bolt.	Nut Across Flats.									Set-Screw.		Dowel.		
		A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*K*	*L*	*M*	*N*	*P*
1/2	7/8	3/4	13/16	1 1/4	1	7/16	9/16	17/32	1/4	1/4	9/32	1/8	1/8	1/8
5/8	1 1/16	15/16	1	1 5/8	1 1/16	7/16	5/8	19/32	1/4	1/4	5/16	1/8	1/8	1/8
3/4	1 1/4	1 1/8	1 3/16	1 7/8	1 3/16	9/16	5/8	19/32	1/4	1/4	5/16	3/16	3/16	3/16
7/8	1 7/16	1 1/4	1 5/16	2 1/4	1 3/8	5/8	3/4	23/32	5/16	5/16	5/8	3/16	3/16	3/16
1	1 5/8	1 7/16	1 1/2	2 1/2	1 1/2	5/8	7/8	27/32	3/8	3/8	7/16	1/4	1/4	1/4
1 1/8	1 13/16	1 9/16	1 11/16	2 3/4	1 11/16	13/16	7/8	27/32	3/8	3/8	7/16	5/16	5/16	5/16
1 1/4	2	1 3/4	1 7/8	3	1 13/16	15/16	7/8	27/32	3/8	3/8	7/16	5/16	5/16	5/16
1 3/8	2 3/16	1 15/16	2 1/16	3 1/4	2	1 1/8	7/8	27/32	3/8	3/8	7/16	5/16	5/16	5/16
1 1/2	2 3/8	2 1/8	2 1/4	3 1/2	2 1/8	1	1 1/8	1 3/32	1/2	1/2	9/16	3/8	3/8	3/8
1 5/8	2 9/16	2 5/16	2 7/16	3 3/4	2 5/16	1 3/16	1 1/8	1 3/32	1/2	1/2	9/16	3/8	3/8	3/8
1 3/4	2 3/4	2 1/2	2 5/8	4	2 7/16	1 5/16	1 1/8	1 3/32	1/2	1/2	9/16	3/8	3/8	3/8
1 7/8	2 15/16	2 11/16	2 13/16	4 1/4	2 9/16	1 7/16	1 1/8	1 3/32	1/2	1/2	9/16	3/8	3/8	3/8
2	3 1/8	2 7/8	3	4 1/2	2 3/4	1 3/8	1 3/8	1 5/16	1/2	1/2	11/16	3/8	3/8	3/8
2 1/8	3 5/16	3 1/16	3 3/16	4 3/4	2 15/16	1 9/16	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 1/4	3 1/2	3 1/4	3 3/8	5	3 1/16	1 11/16	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 3/8	3 11/16	3 7/16	3 9/16	5 1/4	3 3/16	1 13/16	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 1/2	3 7/8	3 5/8	3 3/4	5 1/2	3 3/8	2	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 5/8	4 1/16	3 13/16	3 15/16	5 3/4	3 1/2	2 1/8	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 3/4	4 1/4	4	4 1/8	6	3 11/16	2 5/16	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
2 7/8	4 7/16	4 3/16	4 5/16	6 1/4	3 13/16	2 7/16	1 3/8	1 5/16	5/8	5/8	11/16	3/8	3/8	3/8
3	4 5/8	4 3/8	4 1/2	6 1/2	4	2 9/16	1 5/8	1 9/16	3/4	3/4	13/16	7/16	7/16	7/16
3 1/4	5	4 3/4	4 7/8	7	4 5/16	2 11/16	1 5/8	1 9/16	3/4	3/4	13/16	7/16	7/16	7/16
3 1/2	5 3/8	5 1/8	5 1/4	7 1/2	4 5/8	3	1 5/8	1 9/16	3/4	3/4	13/16	7/16	7/16	7/16
4	6 1/8	5 7/8	6	8 1/2	5 1/4	3 5/8	1 5/8	1 9/16	3/4	3/4	13/16	7/16	7/16	7/16

4. Self-Locking Threads. — In the "Harvey Grip" thread, the bolt has a ratchet-thread, under cut on the bearing side at about 5 degrees *less* than a right angle to the axis of the bolt and the apex of the thread is cut to a knife-edge. The nut also has a ratchet-thread, the bearing side of which is about 5 degrees *greater* than a right angle to the axis of the nut. There is thus a cavity

of about 10 degrees between the bolt and nut-threads; and, when the nut is screwed home, the axial pressure upon it forces the thin bolt-threads out into the nut-threads, thus filling the cavity and locking the nut.

In another locking device of this class, the thread is triangular with the V cut off at $\frac{1}{3}$ of its height from the top and filled in at $\frac{1}{8}$ the height from the bottom. The thread is thus about $\frac{1}{2}$ the height of the sharp V type and has broad flats. The threaded portion of the bolt has a taper of 1 in 48 to the axis, while the nut has the usual thread and is tapped straight. Hence, as the latter moves up the conical surface of the bolt, the metal of the broad-topped bolt-threads flows into the narrower nut-spaces causing the threads to lock tightly. The fibre of the metal displaced in screwing the nut on, is broken when the nut is unscrewed.

5. Spring-Washers. — A spring-washer, such as is shown in Fig. 52*a*, is used frequently as a locking device. It is, in effect, one convolution of a helical spring which is interposed between the nut and the member to be secured. The nut is screwed home upon the washer and the elasticity of the latter produces a pressure upon the nut and, therefore, increased frictional resistance of the threads.

a *b* *c*

Fig. 52.

Fig. 52*b* represents one form of the Verona nut-lock, a spring washer which is not curved helically as a whole, but has the points thrown out, thus giving added power and cutting edges which engage the abutting surfaces. The tail-piece extension is used in railway construction in keeping the lock clear of the oval holes punched in fish-plates.

The National Lock Washer, shown in Fig. 52*c*, has a sharp rib on its inner circumference and next the nut-face. When the nut is set up, it meets the rib, which, being harder than the nut, progressively upsets and forces some of the metal of the latter into the bolt-threads. Hence, the nut is held not only by spring pressure but by a partially locked nut-thread.

Fig. 53 illustrates the Excelsior Double Nut Lock as applied to a fish-plate. It is of serpentine form with two loops and out-thrown points, and is bent into a shallow elliptic curve. Since it embraces two bolts, it cannot rotate with the nuts.

FIG. 53.

6. LOCK-PLATES. — The nut may be kept from reversing by a lock-plate fastened at one side of it, as in Fig. 54. The plate is held by a cap-screw tapped into the flange to be secured and is essentially but a thin wrench engaging the nut. The form shown will hold the nut in either of two positions, *i. e.*, with a side of the nut parallel or perpendicular to the centre-line, *B-C*. Lock-plates, single or double, are used frequently for the nuts of studs which join propeller-blades to the hub. The plate shown in Fig. 54 may have a slot for the screw, *C*, concentric with the bolt-centre. In that case, the plate may be shifted and the nut locked in any position.

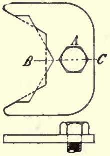


FIG. 54.

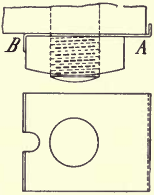


FIG. 55.

The Jones Tie-Bar Lock is shown in Fig. 55. It is a square washer with one end, *A*, flanged upward against the bar and the other extremity, *B*, bent downward against the side of the nut. The latter flange, *B*, is turned after the nut is in place.

7. SPLIT PINS. — Nuts not requiring frequent removal, as those of piston follower-bolts, are sometimes fitted with split pins. After the nut is in place, a hole is drilled through the bolt so that the pin when inserted will bear upon, and prevent axial movement of, the nut. Such a lock serves for but one position of the parts.

34. Wrenches.

1. U. S. Standard. — Table XXXVII. and Fig. 56 — which are reproduced herein through the courtesy of Messrs. J. H. Williams and Company, Brooklyn, N. Y. — give the proportions of Engineers' Wrenches, Single Head, drop-forged, for the nuts of bolts ranging in diameter from $\frac{1}{8}$ inch to $3\frac{1}{2}$ inches.

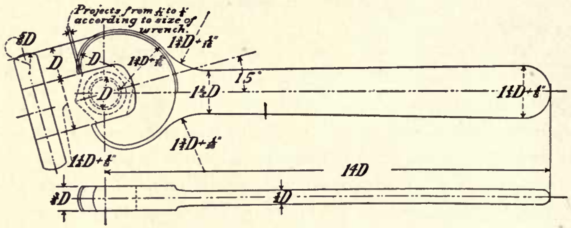


Fig. 56.

Table XXXVII.

Engineers' Wrenches, Single Head.

(Messrs. J. H. Williams & Co.)

Number.	For U. S. Standard Nut; Size Bolt.	Opening Finished.	Extreme Length.	Thickness Head.
00	$\frac{1}{8}$	$\frac{5}{16}$	$2\frac{1}{2}$	$\frac{3}{16}$
0	$\frac{3}{16}$	$\frac{13}{32}$	$2\frac{7}{8}$	$\frac{7}{32}$
1	$\frac{1}{4}$	$\frac{1}{2}$	$3\frac{3}{4}$	$\frac{1}{4}$
2	$\frac{5}{16}$	$\frac{19}{32}$	$4\frac{3}{4}$	$\frac{9}{32}$
3	$\frac{3}{8}$	$\frac{11}{16}$	$5\frac{5}{8}$	$\frac{5}{16}$
4	$\frac{7}{16}$	$\frac{25}{32}$	$6\frac{1}{2}$	$\frac{11}{32}$
5	$\frac{1}{2}$	$\frac{7}{8}$	$7\frac{1}{2}$	$\frac{25}{64}$
6	$\frac{9}{16}$	$\frac{31}{32}$	$8\frac{3}{8}$	$\frac{7}{16}$
7	$\frac{5}{8}$	$1\frac{1}{16}$	$9\frac{1}{4}$	$\frac{31}{64}$
8	$\frac{3}{4}$	$1\frac{1}{4}$	$11\frac{1}{8}$	$\frac{9}{16}$
9	$\frac{7}{8}$	$1\frac{7}{16}$	13	$\frac{21}{32}$
10	1	$1\frac{5}{8}$	$14\frac{7}{8}$	$\frac{3}{4}$
11	$1\frac{1}{8}$	$1\frac{13}{16}$	$16\frac{5}{8}$	$\frac{53}{64}$
12	$1\frac{1}{4}$	2	$18\frac{1}{2}$	$\frac{29}{32}$
13	$1\frac{3}{8}$	$2\frac{3}{16}$	$20\frac{1}{4}$	$\frac{63}{64}$
14	$1\frac{1}{2}$	$2\frac{3}{8}$	$22\frac{1}{4}$	$1\frac{1}{16}$
15	$1\frac{5}{8}$	$2\frac{9}{16}$	24	$1\frac{1}{8}$
16	$1\frac{3}{4}$	$2\frac{3}{4}$	$25\frac{3}{8}$	$1\frac{7}{32}$
$16\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{15}{16}$	$25\frac{3}{8}$	$1\frac{7}{32}$
17	2	$3\frac{1}{8}$	$29\frac{1}{2}$	$1\frac{3}{8}$
18	$2\frac{1}{4}$	$3\frac{1}{2}$	33	$1\frac{17}{32}$
19	$2\frac{1}{2}$	$3\frac{7}{8}$	37	$1\frac{5}{8}$
$19\frac{1}{2}$	$2\frac{3}{4}$	$4\frac{1}{4}$	37	$1\frac{5}{8}$
20	3	$4\frac{5}{8}$	44	$1\frac{7}{8}$
$20\frac{1}{2}$	$3\frac{1}{2}$	$5\frac{3}{8}$	44	$1\frac{7}{8}$

The length and thickness of similar wrenches — excepting that the handle tapers in the opposite direction — are given as, respectively, 59 inches and $2\frac{5}{8}$ inches. It will be observed that the opening of these wrenches is at an angle of 15 degrees with the handle. This inclination permits the turning of a hexagon nut completely around in positions where the swing of the handle is limited to 30 degrees — an important improvement which originated with this firm. The proportions given in Fig. 56 are those of a wrench of medium size, the unit being the bolt-diameter. These proportions are modified somewhat as the wrenches become very large or very small, although the general design remains the same. Check-nut wrenches are shorter and, of course, thinner. Their dimensions are given in Table XXXVIII.

TABLE XXXVIII.

CHECK-NUT WRENCHES.

(MESSRS. J. H. WILLIAMS & CO.)

Number.	For U. S. Standard Nut; Size Bolt.	Opening, Finished.	Extreme Length.	Thickness Head.
602	$\frac{5}{16}$	$\frac{19}{32}$	$4\frac{1}{4}$	$\frac{11}{64}$
603	$\frac{3}{8}$	$\frac{11}{16}$	$5\frac{1}{8}$	$\frac{3}{16}$
604	$\frac{7}{16}$	$\frac{25}{32}$	$5\frac{7}{8}$	$\frac{7}{32}$
605	$\frac{1}{2}$	$\frac{7}{8}$	$6\frac{5}{8}$	$\frac{1}{4}$
607	$\frac{5}{8}$	$1\frac{1}{16}$	$8\frac{1}{4}$	$\frac{5}{16}$
608	$\frac{3}{4}$	$1\frac{1}{4}$	10	$\frac{3}{8}$
609	$\frac{7}{8}$	$1\frac{7}{16}$	$11\frac{1}{2}$	$\frac{7}{16}$
610	1	$1\frac{5}{8}$	$13\frac{1}{4}$	$\frac{1}{2}$

2. INTERNATIONAL STANDARD THREAD (S. I.). — The origin and proportions of this system of screw-threads have been described in § 20. A special committee of delegates from the Association of German Engineers, the Society for the Encouragement of National Industry at Paris, and the Swiss Union of Mechanical Manufacturers met at Zurich, October 20, 1900, to formulate an auxiliary standard system of wrench openings for nuts and bolt-heads. A conference of delegates from these societies, on October 30, 1900, adopted and recommended for international use the system whose rules * follow:

The standard openings are considered as limiting dimensions which the nut is not to exceed nor the wrench fall short of.

To each diameter of the standard series corresponds a particular wrench opening.

* *American Machinist*, April 4, 1901.

The same openings should be employed for diameters specially intercalated, between the standard ones. (This evidently means that where a bolt of special diameter is made, it should be given a head and nut of a standard size.)

The opening of the wrench is the same for the nut and for the head of the bolt and the screw of the same diameter.

The same opening is applicable to rough nuts and machined nuts.

It is recommended that the height of the nut be equal to the diameter, and of the head to seven tenths of the diameter.

The following table gives these openings for all the standard diameters :

Diameter of the Screw.	Pitch.	Opening of the Wrench.	Diameter of the Screw.	Pitch.	Opening of the Wrench.
mm.	*mm.*	*mm.*	*mm.*	*mm.*	*mm.*
6	1	12	33	3.5	50
7	1	13	36	4	54
8	1.25	15	39	4	58
9	1.25	16	42	4.5	63
10	1.5	18	45	4.5	67
11	1.5	19	48	5	71
12	1.75	21	52	5	77
14	2	23	56	5.5	82
16	2	26	60	5.5	88
18	2.5	29	64	6	94
20	2.5	32	68	6	100
22	2.5	35	72	6.5	105
24	3	38	76	6.5	110
27	3	42	80	7	116
30	3.5	46			

The wrench openings in the above table approximate those deduced from the formula 1.4 diameter (in millimeters) $+ 4$ mm.

CHAPTER III.

RIVETED JOINTS. THEORY AND FORMULÆ.

35. Rivets.

RIVETS are permanent fastenings used in joining the parts of metallic structures, such as the framing of buildings and bridges, the hulls of ships, the shells of steam boilers, and the plating of tanks, gasometers, etc. They are made in a forging machine (§ 31), the dies of which form under pressure, from the heated bar, a rivet-blank composed of the *head* and the *shank* or body. When the blank is reheated and set in the joint, a second head or *point* is made by hand or power from the metal of the protruding extremity of the shank.

36. Proportions of Rivets.

1. HEAD AND POINT. — The shape of the head is usually spherical or that of a frustum of a cone; that of the point may be spherical, conoidal, or conical, the latter being the usual form with hand-work. Either the head or point or both may be countersunk and recessed in the plate, having then the form of an inverted, conical frustum.

In the United States, riveted joints are designed without regard to the resistance to yielding opposed by the friction between the plates, the rivet being assumed to have practically the same function as that of a bolt subjected only to cross-shear. In effect, however, the rivet has an initial tension due to its contraction in cooling; and, further, from the same cause, the shank is smaller than the hole through which it passes. Therefore, when the joint is loaded, bending stress precedes shearing in the rivet, and, in service, the latter thus meets compound stress of which tension is a factor.

While, therefore, the rivet is not intended for, and is untrustworthy in, tension, that stress acts in service within the shank, producing a consequent compression and tendency to shear within the head and point and to rupture at the junctions of these features with the shank, especially if slight fillets are not made at these places. In experiments made by Stoney on the

strength of iron rivets in tension, he found that, with $\frac{3}{4}$-inch rivets with pan heads and hand-made snap-points, in punched holes, the heads or points flew off under an average tensile stress of 12.32 tons per sq. in. of rivet cross-section. It is apparent that the contour and strength of these features of the rivet are important, not only because of the stresses met, but, as in marine work, bridges, etc., where minimum weight is desirable.

Good practice, with regard to the proportions of rivet-blanks for general service, is given by Table XXXIX. and Fig. 57, which

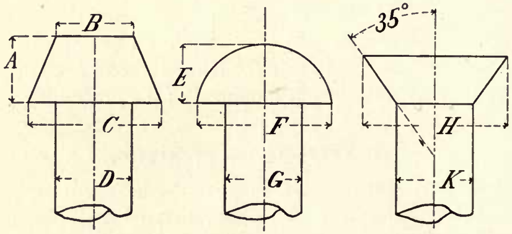


FIG. 57.

illustrate cone or "pan-head," spherical or "button-head," and countersunk-head types, as designed by J. H. Sternbergh, Esq., President of the American Iron and Steel Manufacturing Company.

TABLE XXXIX.

PROPORTIONS OF RIVET-HEADS.

(AMERICAN IRON AND STEEL MANUFACTURING COMPANY.)

Shank, Diameter.	Head.				
	Form.	Diameter, Least.	Diameter, Greatest.	Height.	Angle.
D	Cone.	$B = D$	$C = 1.75\ D$	$A = .875\ D$	
G	Button.		$F = 1.75\ G$	$E = 0.75\ G$	
K	Countersunk.		H		35°

K	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
H	$\frac{1}{2}$	$\frac{19}{32}$	$\frac{11}{16}$	$\frac{25}{32}$	$\frac{7}{8}$	$\frac{31}{32}$	$1\frac{1}{16}$	$1\frac{5}{32}$	$1\frac{1}{4}$	$1\frac{11}{32}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{13}{16}$	2

Similar proportions of Victor Steel Rivets, as made by the Champion Rivet Company, are shown in Table XL. and Fig. 58.

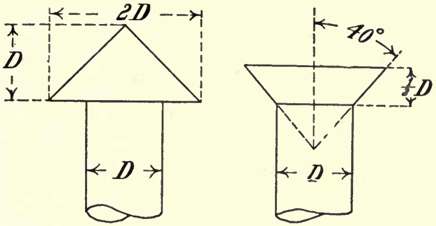


Fig. 58.

The *button-head* form, Fig. 59*a*, is widely used for points and, in structural work especially, for heads as well, excepting where, from lack of space, the countersunk type, Fig. 59*b*, is required.

TABLE XL.

PROPORTIONS OF RIVET-HEADS.
(CHAMPION RIVET COMPANY.)

Shank, Diameter.	Head.				
	Form.	Diam. Least.	Diam. Greatest.	Height.	Angle.
D	Cone.	$\frac{15}{16}D$	$1\frac{3}{4}D$	$\frac{7}{8}D$	
D	Button.		$1\frac{3}{4}D$	$\frac{3}{4}D$	
D	Steeple.		$2\ D$	D	
D	Countersunk.			$\frac{1}{2}D$	40°

The former is much more trustworthy than the conical or *steeple* point, Fig. 59*c*, usual with hand-work.

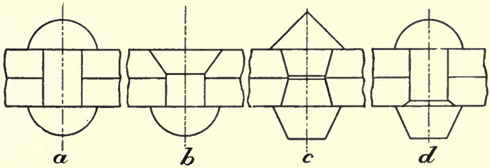


Fig. 59.

The *pan-head*, Fig. 59, *c*, *d*, is much employed in boiler and marine work generally. The form is one of great strength. The objections to it are its weight and the fact that unless its shortest diameter is equal to that of the rivet-shank, it is difficult to make the latter and the head concentric.

The *countersunk* head or point, Fig. 59*b*, adds no weight to the joint and, when properly closed, its wedge-like form gives rigidity and produces maximum plate-friction. There is, however, a decrease in the strength of the plate, owing to the additional metal removed and an increase in cost from the countersinking required. It is essential that countersunk heads shall fit the holes exactly when the rivet is driven home. The angle of countersink varies from 15 to 45 degrees.

2. SHANK. — The shank is cylindrical throughout the greater part of its length but tapers slightly toward the end. Its length is equal to the *grip* (*i. e.*, the combined thicknesses of the plates through which it passes) plus that of the additional metal required to fill the rivet-hole and to form the point. To permit the insertion of the heated and expanded rivet-blank, the rivet holes are usually $\frac{1}{16}$ in. larger in diameter than the blank when cold. Again, in machine-riveting, the pressure upon the hot and plastic metal of the rivet is more continuous and severe than in hand-work, thus forcing more metal into the hole.

Hence, the required length of shank, additional to the grip, depends upon the form of the head, the length and clearance of the rivet-hole, and the character of the riveting process. In average proportions, the total length of the shank is equal to the grip plus 1.5 times the diameter, with an increase, fixed by experiment, for machine-riveting. The length of shank required to make a countersunk point is about that of the shank-diameter.

The slight tapering of the shank under the head, Fig. 59*d*, adds strength at their junction and gives a better form for the conical hole made in punching. In drilled holes a short countersink is advantageous at this point in removing sharp edges left by the tool.

3. RIVET-HEADS AND PLATE-FRICTION. — The results of Professor Bach's extensive experiments upon plate-friction will be given in § 46. With regard to the magnitude of the friction between the plates produced by the pressure of different forms of rivet-heads, Stoney * gives the following results from tests made with steel rivets and steel plates, the joint being composed of a middle plate between two others, all united by three rivets in one row, the rivets being thus in double shear. The 1-inch rivets

* "Strength and Proportions of Riveted Joints," 1885, p. 75.

were used in ¾-inch plates and the ¾-inch rivets in ½-inch plates. With *hand*-riveting, the mean frictional resistances, per rivet in tons, were:

	1-in. Rivet.	¾-in. Rivet.
Snap head and point	6.40	4.72
Pan head, boiler point	7.36	4.52
Pan head, countersunk point	8.55	6.25
Countersunk head and point	9.04	4.95

As a whole, these tests show the greatest friction for countersunk rivets, whose wedge-shaped heads, when properly driven, produce great pressure as the rivet contracts. In other experiments with snap-heads and points, but with *machine*-riveting, the mean friction per rivet was 9.6 tons for 1-inch rivets and 5.9 tons for ¾-inch rivets.

37. Rivet and Plate Metals.

1. Steel has very largely superseded wrought iron for rivets, plating, shapes, etc., in all structural, ship, and boiler-work. The following extracts, with regard to chemical and physical properties, are taken from "The American Standard Specifications for Steel,"* adopted August, 1901, by the American Section of the International Association for Testing Materials:

Structural Steel for Buildings.

1. Steel may be made by either the open-hearth or Bessemer process.

2. Each of the two classes of structural steel for buildings shall not contain more than 0.10 per cent. of phosphorus.

3. There shall be two classes of structural steel for buildings, namely: Rivet Steel and Medium Steel, which shall conform to the following physical qualities:

	Rivet Steel.	Medium Steel.
Tensile strength, lbs. per sq. inch.	50,000–60,000	60,000–70,000
Yield point, in lbs. per sq. in., shall not be less than	½ T. S.	½ T. S.
Elongation, per cent. in eight ins., shall not be less than	26	22

Structural Steel for Bridges and Ships.

1. Steel shall be made by the open-hearth process.

2. Each of the three classes of structural steel for bridges and ships shall conform to the following limits in chemical composition:

	Steel Made by the Acid Process. Per cent.	Steel Made by the Basic Process. Per cent.
Phosphorus shall not exceed	0.08	0.06
Sulphur shall not exceed	0.06	0.06

* "American Standard Specifications for Steel," A. L. Colby, 1902.

3. There shall be three classes of structural steel for bridges and ships namely: RIVET SEEEL, SOFT STEEL, and MEDIUM STEEL, which shall conform to the following physical qualities:

	Rivet Sieel.	Soft Steel.	Medium Steel.
Tensile strength, lbs. per sq. in.	50,000–60,000	52,000–62,000	60,000–70,000
Yield point, in lbs. per sq. in., shall not be less than	½ T. S.	½ T. S.	½ T. S.
Elongation, per cent. in eight inches, shall not be less than	26	25	22

OPEN-HEARTH BOILER PLATE AND RIVET STEEL.

1. Steel shall be made by the open-hearth process.

2. There shall be three classes of open-hearth boiler-plate and rivet-steel, namely: FLANGE OR BOILER STEEL, FIRE-BOX STEEL, and EXTRA SOFT STEEL, which shall conform to the following limits in chemical composition:

	Flange or Boiler Steel. Per cent.	Fire-Box Steel. Per cent.	Extra Soft Steel. Per cent.
Phosphorus shall not exceed	Acid, 0.06; Basic, 0.04	Acid, 0.04; Basic, 0.03	0.04
Sulphur shall not exceed	0.05	0.04	0.04
Manganese.	0.30 to 0.60	0.30 to 0.50	0.30 to 0.50

4. The three classes of open-hearth boiler-plate and rivet-steel, namely: FLANGE OR BOILER STEEL, FIRE-BOX STEEL, and EXTRA SOFT STEEL, shall conform to the following physical qualities:

	Flange or Boiler Steel.	Fire-Box Steel.	Extra Soft Steel.
Tensile strength, lbs. per sq. in.	55,000–65,000	52,000–62,000	45,000–55,000
Yield point, in lbs., per sq. in., shall not be less than	½ T. S.	½ T. S.	½ T. S.
Elongation, per cent. in eight inches, shall not be less than	25	26	28

In all of the steels described above, modifications are made, for thin and thick material, in the required elongation.

In general, steel rivets should be made by the open-hearth process, be low in sulphur and phosphorus, and be of a soft, ductile character. The following table gives the average of a number of analyses of Victor Steel Rivets:

Phosphorus, per cent.	0.015
Manganese, " "	0.460
Sulphur, " "	0.032
Silicon, " "	0.005
Carbon, " "	0.110

With steel, there is practically no change in tenacity when tested with or across the direction of rolling. The results of numerous

experiments indicate that the *ultimate shearing strength* of mild steel may be taken generally as 80 per cent. of the ultimate tensile strength. The *allowable bearing stress* upon the rivet or the surrounding metal ranges usually from 12,000 lbs. to 24,000 lbs. per square inch of the projected semi-intrados (diameter × thickness), although considerable latitude is given this stress by various designers.

2. Wrought Iron. — Iron plates, rods, etc., differ widely in quality, owing to the nature of the processes through which the material passes in manufacture. In the puddling furnace, there appear globules of wrought iron whose centres consist of excess carbon and impurities. These, when passed through the rolls, are stretched into fibres whose outer surfaces are of soft iron, while the interiors contain foreign material as above. As a result of this lack of homogeneity, the fracture in some cases appears fibrous; in others, from 30 to 40 per cent. crystalline.

The *ultimate tensile strength with the grain, i. e.*, parallel to the direction of rolling, ranges from 45,000 to 55,000 lbs. per square inch. *Across the grain*, this strength is less, being, according to Bauschinger's experiments, about 78 per cent. of that with the grain.

With regard to the *shearing strength* of wrought iron, Professor J. B. Johnson * gives the following summary of Bauschinger's elaborate experiments:

"In general, we may say that the shearing strength across the thickness of the plate, either with or across the grain, is about 80 per cent. of the tensile strength, while, if the external forces lie in the plane of the plate and be applied on the planes of shear perpendicular to the plane of the plate, the shearing strength is about the same as the tensile strength. The shearing resistance on a plane parallel to the plane of the plate, is less than 45 per cent. of the tensile strength."

The allowable bearing stress for pins and rivets upon the surface of the projected semi-intrados is usually taken in structural work as 12,000 pounds per square inch.

Despite the widespread introduction of steel, the use of wrought-iron rivets still finds favor, especially in locomotive work. It is stated that, for the sizes used in locomotives, steel rivets, machine-driven, are not so trustworthy as first-class iron rivets, the reason given, being that:

"A rapid distortion at one operation of the steel formed head is more than liable to reduce the tensile strength of the head. In other words, were the steel rivet driven by

* "Materials of Construction," 1898, p. 486.

hand, the head would be stronger than when driven by machine and the contrary would be the case with the iron rivet. This is well recognized in conditions where snap-riveting is required and a leakage of the rivet in service requires calking. Under these conditions the steel rivet will stand more calking than the iron rivet, for the reason that the working due to hard driving has a refining effect on the steel and seems to improve its toughness, whereas the distortion and twisting of the grain of the iron rivet in driving, seems to weaken instead of strengthen it." *

3. SHEARING STRENGTH OF RIVETED JOINTS. — For iron rivets in steel plates, Traill † gives $\frac{8}{13}$ as the ratio, in single shear, between the mean shearing strength per sq. in. of the rivet and the mean tensile strength per sq. in. of the plate. For steel rivets in steel plates, this ratio becomes $\frac{23}{28}$. A rivet in double shear he assumes to have 1.75 times its strength in single shear. Mr. J. M. Allen ‡ takes 38,000 lbs. per sq. in. as the strength in single shear of an iron rivet in steel plates and assumes, in double shear, an increase of 85 per cent., or a total strength of 70,300 lbs. per sq. in.

4. COPPER, when used for the fire-box plates or stay-bolts of locomotive boilers, should have a minimum tensile strength of 30,000 lbs. per sq. in. and an elongation of at least 20 per cent. in a section originally 2 ins. long.

38. Rivet-Holes.

1. MODERN PRACTICE, as to punching or drilling, varies somewhat, although the tendency toward the drilled hole, with its greater accuracy and small liability to injury of the metal, grows steadily. In *boiler-work*, the U. S. Naval specifications require all rivet-holes to be drilled with the plates in position. The rules of the American Boiler Manufacturers' Association permit punched holes in steel plate up to ⅝ inch thick ; in thicker plate, the holes may be either drilled or be punched and reamed. In *structural-work* the holes are punched, as a rule. For field-rivets, they are drilled to templet or reamed with the connected parts in place. In *hull-work*, rivet-holes are generally punched from the faying surfaces of the parts to be connected.

2. ULTIMATE TENSILE STRENGTH OF PERFORATED PLATES. — If a plate be perforated with a row of holes, as for riveting, by methods, as drilling, which produce no molecular disturbance within the metal immediately surrounding the hole, leaving that

* *Am. Engineer, Car Builder, and Railroad Journal*, May, 1898.

† "Boilers: Marine and Land," 1896, pp. 44, 45.

‡ Sibley College Lectures, 1890–1.

metal unchanged in structure, the plate will break, when tested to destruction, in the line of the reduced section remaining through the rivet-holes; but the ultimate tensile strength of that reduced section will be found to exceed materially that of the unperforated metal. In other words, if two plates of the same dimensions and material be thus treated, one solid throughout, the other perforated as above, they will rupture at different total loads, but the ultimate tensile strength, *per square inch*, of the net section along the line of holes will be greater than that of the metal in the solid plate. This apparent paradox is analogous to that which occurs with the "grooved specimens" discussed in § 27. The reduction in sectional area of metal along the holes lessens the space for the flow of that metal and checks its tendency to stretch. Hence, the contraction of area is hindered and opposition to contraction gives increase in tensile strength. When the holes are punched, the specimen is still "grooved" in type, but the condition that the metal surrounding the hole shall be uninjured by the perforation, holds no longer. Therefore, the gain in tensile strength is, in very thin plates, nullified; and, in thicker plates, reduced by the loss in quality of the material.

3. PUNCHES AND DIES. — In punching, as shown in Fig. 60, the plate rests on a die the bore of which is conical with the smaller diameter toward the punch. The base of the latter may be flat, giving a full circumference of cutting surface in action on contact, or the cutting edge may be slanting or spiral (Fig. 60*a*) so that, on contact, only part of the circumference is cutting and, for a time, shearing proceeds in detail.

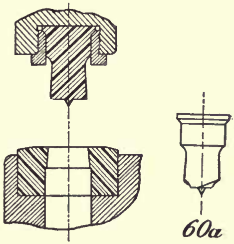


FIG. 60.

With the spiral form, there is a saving in power when the thickness of the plate is less than $\frac{2}{3}$ the diameter of the hole to be punched. For plates beyond that thickness, the flat-faced punch is better. In order to reduce the stress in the plate-metal, the die is made conical, as above, and its diameter is greater than that of the punch by 10 to 15 per cent. The general practice is to make the diameter of the die equal to that of the punch, plus 0.1 to 0.3 times the thickness of plate.

The punch is subjected to crushing stress. The resistance to punching may be taken generally as that of shearing a section equal to the circumference of the hole multiplied by the thickness of the plate. Since a punch cannot withstand more than the total crushing force corresponding with its cross-section, it is apparent that there is a fixed limit to the thickness of plate which it will pierce.

4. Effects of Punching. — On contact, the punch shears the circumference of the blank to be removed, thrusting, in its advance, upon the body of the latter so that there is not only detrusion but a lateral, plastic flow of a portion of the metal of the blank into the walls of the hole. The blank, when ejected, is, as the experiments of Townsend (§ 31) show, no denser than the original plate but its volume is less than that of the hole.

The lateral flow produces molecular disturbance within the metal immediately around the hole, and a portion of this metal becomes dense and hard with a decrease in ductility and rise in elastic limit. There is also a loss in ultimate strength which may possibly arise from minute cracks in the affected metal. Since the thickness of the plate determines the allowable pressure upon the punch, the injury is less with thin plates. It is also smaller with ductile material, mild steel being stressed less than wrought iron.

Some experiments indicate that the flow and hardening are greater on the die side of the plate, while others show that the affected zone lies nearer the upper surface. In any event, the injured metal appears to be included wholly within an annular cylinder, $\frac{1}{16}$ inch or less in thickness around the hole. The remedy, therefore, is to punch the hole $\frac{1}{8}$ inch smaller in diameter than desired and ream to finished size or else to anneal the plate. Either of these methods will remove the ill effects of punching.

The loss of tenacity in punched plates not subsequently reamed or annealed, is with plates $\frac{1}{2}$ inch thick and upward, from 10 to 25 per cent. in iron plates and from 10 to 35 per cent. in steel, the loss in the latter increasing with the thickness. The excess tenacity of a drilled plate is usually 10 to 12 per cent., although its maximum range may be double this, since this gain depends upon the proportions of the "grooved" specimen and the character of the metal.

5. Drilled Holes have none of the disadvantages of those made by punching. The metal about the hole is uninjured since there is little pressure upon it and no lateral flow exists. The blank is removed from the hole in detail by cutting instead of be-

ing forced out bodily by pressure and shearing. The drilled hole should be slightly countersunk to remove the sharp edge, which, when the joint is loaded, would aid in shearing the rivet.

6. TESTS OF DRILLED AND PUNCHED PLATES. — The number of such tests with joints, is large. The following tables * give, in summary, the results of experiments by Mr. Kirkaldy to determine the ultimate tensile strength of steel plates: (*a*) drilled; (*b*) punched; (*c*) punched and afterward annealed. Plates 12 inches wide were used. In the $\frac{1}{4}$-inch, $\frac{1}{2}$-inch, and a part of the $\frac{3}{4}$-inch thicknesses, the number and diameter of the holes in each half of the specimen were, respectively, 6 inches and 0.79 inch. In the remainder of the $\frac{3}{4}$-inch and in the 1-inch plate, the number and diameter were 6 inches and 1.08 inches, respectively. The results were:

(*A*) *Ultimate Stress per square inch of Gross Area at Holes.* The stresses are given in tons and are calculated with reference to the total sectional area of the plate, including therein the part removed by perforation:

Thickness,	$\frac{1}{4}$ in.	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	1 in.
Drilled,	21.90	19.60	19.65	18.30
Punched,	19.30	16.65	15.80	13.45
" and annealed,	20.15	18.55	18.70	17.80

(*B*) *Mean Stress in tons per sq. in. of Net Section between Holes:*

Thickness,	$\frac{1}{4}$ in.	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	1 in.
Drilled,	36.21	32.44	31.64	29.42
Punched,	31.94	27.53	24.60	21.02
" and annealed,	33.41	30.75	30.05	27.82
Solid Plate,	31.65	29.15	29.70	27.70

(*C*) *Stresses in per cent. per sq. in. of Net Section compared with Solid Plate:*

Thickness,	$\frac{1}{4}$ in.	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	1 in.
Drilled,	113.8	111.1	106.4	106.1
Punched,	101.0	94.2	82.5	75.8
" and annealed,	105.6	105.6	101.0	100.3

(*D*) *Difference in per cent. between the Ultimate Stress per sq. in. of Net Section of Perforated and Unperforated Plates:*

Thickness,	$\frac{1}{4}$ in.	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	1 in.
Drilled,	Gain, 13.8	Gain, 11.1	Gain, 6.4	Gain, 6.1
Punched,	" 1.0	Loss, 5.8	Loss, 17.5	Loss, 24.2
" and annealed,	" 5.6	Gain, 5.6	Gain, 1.0	Gain, 0.3

From (*A*) it will be seen that the punched plates have the least ultimate strength and the drilled plates the greatest. (*D*) shows,

* Merchant Shipping, "Experiments on Steel," 1881, pp. 12–14.

for the drilled plates, a gain in ultimate strength over that of the solid plate of 6.1 to 13.8 per cent., and a loss, in the punched plates from $\frac{1}{2}$-inch upward, of 5.8 to 24.2 per cent. The punched and annealed plates occupy an intermediate position, having a gain which is materially less than that of the drilled plate. The manner in which the ductility of the steel was affected by its treatment is indicated by:

(*E*) *Elongation in per cent. of Holes at Ultimate Stress:*

Thickness,	$\frac{1}{4}$ in.	$\frac{1}{2}$ in.	$\frac{3}{4}$ in.	1 in.
Drilled,	24.3	37.0	37.6	33.5
Punched,	11.7	18.5	11.1	4.3
" and annealed,	27.1	35.1	33.0	29.8

As stated previously, the strength of a grooved specimen depends upon its form, the quality of the metal, and the method of "grooving" or, in these tests, of perforation. Hence the results given apply, quantitatively, only to the specimens tested, although the general principles which are indicated, hold true in all cases.

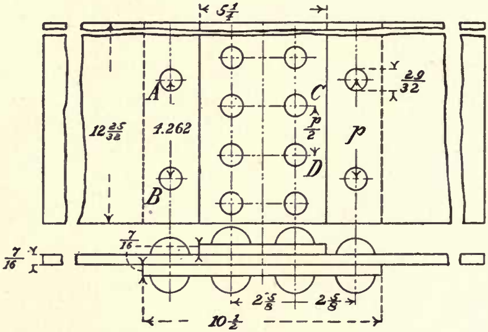


FIG. 61.

7. RIVETED JOINTS WITH PUNCHED OR DRILLED HOLES. — Joints in which the holes are punched or drilled give, under test, similar differences in strength, although the range of variation will not be the same as in the unriveted plates since the joint is a built-up structure and the load is transmitted from one plate to another in a complex manner. For example, in the double-butt-strapped joint shown in Fig. 61,* the plate and straps were $\frac{7}{16}$ inch thick and

* *Jour. Am. Soc. Naval Engineers*, XII., p. 4.

had a tensile strength of 55,000 lbs. per square inch; the rivets were $\frac{29}{32}$ inch diameter, and their strength was 40,000 lbs., and 70,000 lbs. per square inch of section in single and double shear, respectively. The ultimate strengths of the joint, with the rivet holes made as below, were:

	Breaking Load in Lbs.
Holes punched	261,600
" " and reamed	286,800
" drilled	308,200

39. Boiler-Seams: Longitudinal, Circumferential, and Helical.

In the shell of a cylindrical boiler, circumferential or girth seams are perpendicular to longitudinal seams, and the latter are parallel to the axis. Helical, in place of longitudinal, seams have been used to a slight extent for shells, and are employed in riveted pipe. In Fig. 62, let:

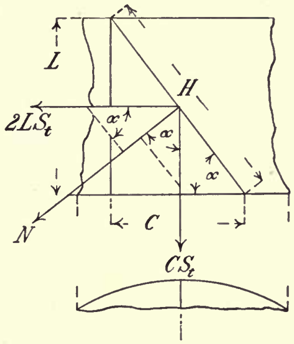


FIG. 62.

$R =$ radius of boiler-shell;
$t =$ thickness of boiler-shell;
$p =$ steam-pressure per gauge;
$C =$ length considered of circumferential seam;
$L =$ length considered of longitudinal seam;
$H =$ length of helical seam equivalent to L;
$S_t =$ unit tensile stress on circumferential seam;
$S_t' =$ unit tensile stress on longitudinal seam;
$S_t'' =$ unit tensile stress on helical seam.

Assume all joints as having the full strength of the plate, *i. e.*, as if welded. The total load on the circumferential seam is that on the boiler-head, or $\pi R^2 \times p$. The resistance of the entire seam is the product of its length, thickness, and the permissible unit stress, or $2\pi R \times t \times S_t$. Equating the load and resistance:

$$S_t = \frac{Rp}{2t}. \tag{77}$$

From equation (4) and Fig. 1:

$$S_t' = \frac{Rp}{t} = 2S_t, \tag{78}$$

i. e., the longitudinal have double the unit stress of the circumferential seams. Expressing C, L and H in the same units:

Normal load on length, $C = C.S_t$;

Normal load on length, $L = L.S_t' = 2L.S_t$.

The normal load, N, on the helical seam is the sum of the components of the loads on seams C and L, which are normal to seam H.

Component of $C.S_t$, normal to $H = C.S_t \cdot \cos\alpha$;

Component of $2L.S_t$, normal to $H = 2L.S_t \cdot \sin\alpha$.

$$N = C.S_t \cdot \cos\alpha + 2L.S_t \sin\alpha.$$

$$H = \sqrt{C^2 + L^2};\ \sin\alpha = \frac{L}{\sqrt{C^2 + L^2}};\ \cos\alpha = \frac{C}{\sqrt{C^2 + L^2}}$$

$$\therefore N = \frac{C^2 + 2L^2}{\sqrt{C^2 + L^2}} \cdot S_t.$$

The unit tensile stress on the helical seam will be equal to the total normal load on the seam divided by the length of the latter, or:

$$S_t'' = \frac{N}{H} = \frac{C^2 + 2L^2}{C^2 + L^2} \cdot S_t. \tag{79}$$

If $C = L$, $S_t'' = 1.5S_t$;

if $C = 2L$, $S_t'' = 1.2S_t$;

if $C = 3L$, $S_t'' = 1.1S_t$.

The stress on the longitudinal seam is $2S_t$ in all cases. It is apparent that the stress on the helical seam decreases as C grows. If $L = 0$, the helical seam becomes circumferential and $S_t'' = S_t$. If $C = 0$, the seam is longitudinal and $S_t'' = 2S_t$.

The strength required in the seam is decreased by the helical form but its cost is increased by the greater length of the joint and by the necessity for the use in all but small boilers of plates with inclined sides, as shown in Fig. 62, laid in circumferential bands or courses. This waste of metal is avoided in Root's spiral riveted pipe, which is made of single strips, joined by welding to any desired length and wound and riveted helically to form the pipe. The thickness of the plate varies from No. 28 to No. 12, B. W. G., and the approximate bursting pressures are given as ranging from 900 to 1,300 lbs. per sq. in. at 3 ins. diameter to 110 to 335 lbs. at 24 ins. diameter. The pipe is used for water, exhaust steam, etc.

40. Forms of Riveted Joints.

The function of a riveted joint may be simply that of resisting direct stresses upon it, as in structural work; or there may be

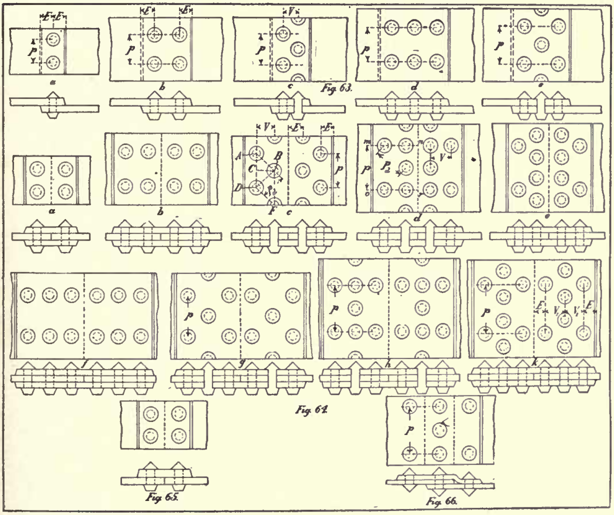


Fig. 63. Fig. 64. Fig. 65. Fig. 66.

added to this the requirement that the joint shall be also tight against fluid pressure. The latter is, in steam boilers, high and internal; in hull and gasometer plating, moderate or light and

external and internal, respectively, to the joint. The duty of the latter affects materially its proportions.

The plates are, in the simplest form of joint, united by being overlapped and riveted; in stronger but more complex forms they abut, the seam being covered infrequently by one, but usually by two, external and internal butt straps or cover plates, which are riveted to the plates and to each other. *Lap Joints* are shown in Fig. 63. One plate rests upon the other and rivets connect them. Fig. 64 illustrates *Double-Strapped Butt Joints.* In this form, the main plates do not overlap, but remain in the same plane, the straps being above and below the latter. Fig. 61 shows a similar joint with straps unequal in width; Fig. 65 a *Single-Strapped Butt Joint*; and Fig. 66 a *Single-Strapped Lap Joint.*

Joints differ also with regard to the number and arrangement of the rows of riveting which are placed parallel to the plate-edges in a lap-joint or on each side of the seam in a butt-joint. There may be from one to four rows, giving a *single, double, treble, or quadruple-riveted joint.* In *chain-riveting* (Fig. 63, *b, d*) the rivets in adjacent rows are set one behind the other on a line perpendicular to the seam. In *staggered (zigzag) riveting* (Fig. 63, *c, e*) the rivets are *en échelon,* being placed on a line which meets the seam at an angle. In both of these forms, alternate rivets in the outer or inner rows or in both may be omitted. *Group riveting,* as shown in Fig. 67, is sometimes employed in structural work. The rivets are disposed usually in arithmetical series, proceeding from the centre outward with an increasing pitch.

41. The Elements of a Riveted Joint.

In order to allow for the expansion of the heated rivet-blank, rivet-holes of average size are made $\frac{1}{16}$ inch larger in diameter than the rivet when cold. In calculating the strength of a joint, the *diameter* of the hole, not that of the unheated rivet-blank, should be considered, since the latter is upset in riveting so that it fills the hole excepting for the slight contraction in cooling. The pitch, *p*, Fig. 63, *a*, is the distance parallel to the seam between consecutive rivets in the same row. Where alternate rivets are omitted in any row, as in Fig. 64, *d*, the pitch of the joint and that used in calculation, is the greatest pitch in the several rows, since lines, as *m-n* and *o-r*, drawn through the centres of its bounding rivets and normal to the seam, will include a section of the

joint which forms a repeating pattern throughout the whole extent of the latter, so that such a section represents fully the construction and strength of the joint.

The *transverse pitch*, or distance between the centre-lines of adjacent rivet-rows in a direction normal to the seam, as V, Fig. 64, *c*, *d*, differs in magnitude in chain and staggered riveting. The *diagonal pitch*, p_d, in the same figures, is the distance between the centre of a rivet and that of the one nearest it diagonally in the

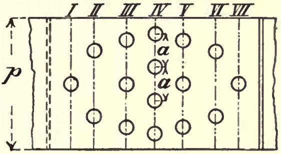


FIG. 67.

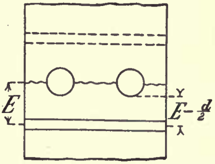


FIG. 68.

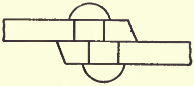


FIG. 69.

next row. The *margin* is the width of plate or strap between the centre of the outer row of rivets and the edge, as E, Figs. 63, *a*, and 64, *c*. The *lap*, in lap-riveting, is the amount by which one plate overlaps the other; in butt-riveting, it is the extent by which the strap overlaps one plate. In both cases it is equal to $2E$, plus the distance between rivet-rows. Before discussing these various elements of the seam in detail, consider

1. THE MANNER OF FAILURE OF A JOINT. — Take the simplest, case — that of the single riveted lap-joint, Fig. 63, *a*. This joint, when tested to destruction, may fail by:

(*a*) Rupturing the plate between the rivet-holes, as in Fig. 68;
(*b*) Shearing the rivets, as in Fig. 69;
(*c*) Rupturing the margin, as in Fig. 70;
(*d*) Shearing the margin, as in Fig. 71;
(*e*) Crushing the plate or rivet, as in Fig. 72.

In staggered riveting, rupture as in (*a*) may proceed along the diagonal pitches, if the latter are weak as compared with the longitudinal pitch. The same action, under the same conditions, may take place in chain-riveting with alternate rivets omitted in the outer row. In staggered riveting with alternate rivets omitted

in the outer row, as in Fig. 73, (*a*) may occur along the pitch, *A-D*, or along two diagonal pitches and the semi-pitch, as *A-B-C-D*. In double butt-strap joints, (*b*) cannot take place unless the main plate shears each rivet at two sections.

In complex joints, failure may be due to both shearing and rupture, thus a lap-joint riveted as in Fig. 64, *h* or *k*, may give way

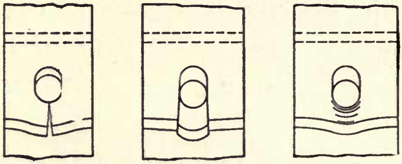


FIG. 70. FIG. 71. FIG. 72.

by shearing the rivets in the outer row and tearing the plate along the rivet-holes of the central row. Each form of joint requires separate investigation with regard to each possible manner of failure. In design, the desire is usually to make the joint, as nearly as possible, equal in strength throughout. Its efficiency is measured by the ratio of the tensile strength of the net section of plate-metal left along the line of the greatest pitch, as compared with that of a similar section, one pitch long, of the solid plate.

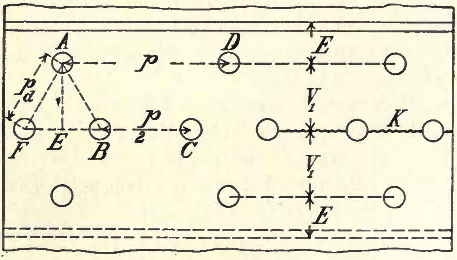


FIG. 73.

An excess of strength, within reasonable limits, in other elements of the joint is not material, if that of the net section as above be fully utilized and yet be slightly less than those of the seam in other respects, so that rupture will occur along that line, since, especially in joints for tightness, an unnecessarily wide pitch not

only gives surplus metal and inequality of strength but adds to the difficulty of making the joint tight.

2. Rivet-Diameter. — In a single-riveted lap-joint, let:

$d =$ diameter of rivet;

$p =$ pitch of rivet;

$t =$ thickness of plate;

$S_t =$ unit ultimate tensile strength of plate;

$S_c =$ unit ultimate crushing strength of plate or rivet;

$S_s =$ unit ultimate shearing strength of rivet.

Then, considering a section of the joint one pitch wide:

Tensile strength, net section of plate $= (p - d)t \cdot S_t$;

Crushing strength, plate or rivet $= d \cdot t \cdot S_c$;

Shearing strength, rivet $= \frac{\pi d^2}{4} \cdot S_s$.

For equality of strength throughout:

$$d \cdot t \cdot S_c = \frac{\pi d^2}{4} \cdot S_s \therefore d = \frac{4}{\pi} \cdot \frac{S_c}{S_s} \cdot t = Ct \qquad (80)$$

$$d \cdot t \cdot S_c = (p - d)t \cdot S_t \therefore d = \frac{S_t}{S_c + S_t} \cdot p = Kp \qquad (81)$$

in which C and K are constants.

It will be seen that, for equal strength throughout, the *maximum* permissible diameter of the rivet is fixed by the thickness of the plate; that, for that maximum diameter, there is but one pitch which is suitable under these conditions; and that, if a less diameter than the maximum be used, the pitch, for equal strength, changes with it.

Again, if the holes are punched, the permissible rivet-diameter for a given thickness of plate is limited also, as stated previously, by the ultimate strength of the punch, which, for crushing, is $\pi d^2/4 \cdot S_c$. The minimum shearing resistance of the plate is the area of the sheared section $\times$ the unit ultimate shearing strength, or $\pi d \cdot t \cdot S_s$. Equating:

$$\frac{\pi d^2}{4} \cdot S_c = \pi d \cdot t \cdot S_s \therefore d = \frac{4S_s}{S_c} \cdot t = k \cdot t,*$$

in which k is a constant.

In structural work, the rivet-diameter is usually $\frac{3}{4}$ in. or $\frac{7}{8}$ in. With cylindrical steam-boilers, the shell-diameter, steam-pressure,

* Unwin: "Machine Design," 1901, I., 122.

type of longitudinal seam, and factor of safety determine the thickness of the shell-sheets. From that thickness, the diameter and pitch for equal strength throughout, may be found from formulæ similar to (80) and (81). In hull-work, the diameter of the rivet varies from $\frac{1}{4}$ in. to $1\frac{1}{4}$ ins., depending upon the thickness of the plating.

3. MULTIPLE RIVETING.—The efficiency of the single-riveted lap joint is but little more than 50 per cent., *i.e.*, $(p - d)t \div p \cdot t = 0.5$, about. Hence, only about one half of the full strength of the connected plates is utilized. This seam is employed only where,

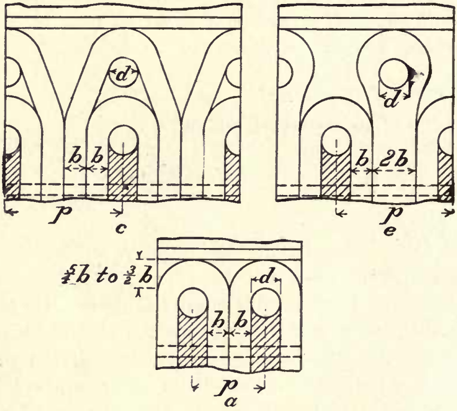


FIG. 74.

owing to caulking, corrosion, or other reasons, a sheet relatively so thick is used that the fractional strength, as above, will suffice to resist the load upon the joint.

Assume a single-riveted lap-joint just capable of bearing, with a proper factor of safety, a given load and let it be desired to augment this load without increasing the thickness of the plate. It is evident that the strength of the joint must be made greater in tension, shearing and bearing to resist stresses (*a*), (*b*), and (*e*), disregarding, for the time, the stresses (*c*) and (*d*) within the margin. To provide for the rise in stress (*a*), the net plate-section must be greater, *i. e.*, there must be a wider pitch. This extended pitch will, from (81) and the increase in shearing load, (*b*), per rivet, necessitate a larger rivet-diameter. The latter, however, is lim-

ited, for equal strength, by (80) and, in practice, by the growth in the pressure required for forming the rivet-point, which pressure increases with the size of the rivet, but is restricted by the thickness of the plate. Again, the pitch, in steam-joints is limited by the necessity for tightness. For these reasons, multiple (double, triple) riveting must be adopted in such a case. There is a marked gain from multiple riveting, owing to the better distribution of the material of the joint.

Graphically, this distribution is shown in Fig. 74, in which the boiler plate is assumed, with regard to shearing and tensile stresses only, to be divided into tension-links and redundant material, the latter being shaded in the diagrams. The width, b, of the link-bars is so proportioned that the total strength of the link in tension, $2b \cdot t \cdot S_t$, is equal to the shearing strength, $\pi d^2/4 \cdot S_s$, of one rivet, the latter being in single shear. To provide for bearing stress, the width of the link at the head should be 1¼ to 1½ times b. In (c) and (e), p and d are the same, the former being relatively greater and the latter relatively less than in (a). Hence, the net-plate-section, $(p - d)t$, and the efficiency, $(p - d)t \div p \cdot t$ are greater in the double-riveted joints and the proportion of redundant material is less.

4. Pitch. — The *minimum* pitch permissible is governed by several considerations. If the distance between adjacent edges of rivet-holes, is less than the diameter of the rivet, the stress in punching or the lateral pressure of the plastic rivet-blank in riveting, may crack the plate between the rivet-holes. Again, the maximum diameter of the rivet-point is usually 1.75 times that of the hole, so that a pitch of two diameters gives barely enough space for the riveting dies. In practice, the minimum pitch is generally from $2.5d$ to $3d$ in various classes of work.

The *minimum* pitch permissible depends upon the service of the joint. If the latter is to be steam-tight, the pitch should be equal to, or less than, that demanded by equality of strength. In such a case, the steam tends to enter between the laps or straps and the plates of a joint; and the strip of metal between two rivets is, very approximately, in the condition of a beam fixed at its ends and loaded uniformly. The maximum deflection of such a beam is:

$$\Delta = \frac{wl^4}{384EI},$$

in which $w=$ unit pressure, $l=$ span, *i. e.*, pitch, $E=$ modulus of elasticity, and $I=$ moment of inertia of the section of the beam. Since the deflection thus varies as the fourth power of the pitch, the latter, to prevent leakage, should be as small as possible.

In structural riveting, this requirement as to fluid pressure does not exist, but the rivets must not be spaced too widely or the joint may open sufficiently for moisture to enter, thus causing rusting and eventually bursting the joint. Again, when the joint is in compression, the strip between two rivets is essentially a column and is, therefore, subject to flexure. In the direction of the stress, the pitch should not exceed, as a rule, 6 inches or 16 times the thinnest outside plate connected. Transversely to the stress, the pitch may be 32 to 40 times that thickness.

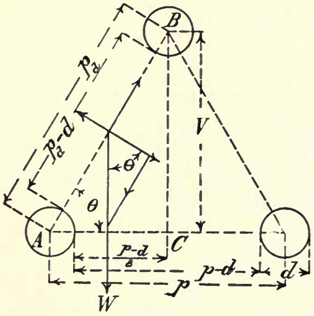


FIG. 75.

5. DIAGONAL PITCH. — In tension-joints, the stress along the line of the pitch, p, is tensile only, while in the direction of the diagonal pitch, p_d, there are both tensile and shearing components. Hence, the resistance of the metal along p_d is, with regard to tension normal to the seam, less than that of the same section if located parallel to p. From Fig. 75,* we have:

Total load on $\frac{1}{2}$ *pitch,* $A\text{-}C = W = (p-d)t \cdot S_t$.

This load must be borne also by the metal along the diagonal pitch, *A-B*. Resolving W parallel and perpendicular to *A-B*:

Shearing component of W along A-B $= (p-d)t \cdot S_t \cdot \sin\theta$;

Tensile component of W, normal to A-B $= (p-d)t \cdot S_t \cdot \cos\theta$.

The unit shearing stress, S_s, on the net section along *A-B* will be equal to the total shearing load on that section, divided by the area of the section, *i. e.*,

$$S_s = \frac{(p-d)t \cdot S_t \cdot \sin\theta}{(p_d-d)t}.$$

* Commander A. B. Canaga, U.S.N., *Jour. Am. Soc. Naval Engrs.*, VIII., 2.

Similarly, the unit tensile stress, S_t', on the net section along A-B will be equal to the total tensile load on that section, divided by the area of the section, *i. e.*,

$$S_t' = \frac{(p-d)t \cdot S_t \cdot \cos\theta}{(p_d - d)t \cdot}.$$

For steel plates, the unit shearing stress on any section should not exceed $\frac{8}{10}$ of the unit tensile stress. Hence:

$$S_s = 0.8\ S_t',$$

$$\therefore \frac{(p-d)t \cdot S_t \cdot \sin\theta}{(p_d - d)t} = \frac{8}{10} \cdot \frac{(p-d)t \cdot S_t \cdot \cos\theta}{(p_d - d)t};$$

$$\sin\theta = \tfrac{8}{10}\cos\theta;\ \tan\theta = 0.8\ \therefore \theta = 38^\circ\ 40'. \quad (82)$$

Also:

$$\tan\theta = V \div \frac{p}{2} = 0.8,\ \therefore\ V = 0.4\,p. \quad (83)$$

The value of V depends thus upon that of θ and the latter is determined by the assumption that $S_s = 0.8\ S_t'$. As a general rule, Traill takes the available resistance of the metal along the diagonal pitch, for tension normal to the longitudinal pitch, as $\frac{5}{6}$ of that of the same section along the latter pitch.

6. Transverse Pitch.—The value, V, of this pitch has been calculated in the preceding section for staggered riveting with no rivets omitted in the outer row (Fig. 64 *c*) and for chain riveting with alternate rivets omitted in the outer row (Fig. 64 *d*). When, in staggered riveting, alternate rivets are omitted in the outer of several rows, the values of V for the outer and the next rows are different, since, as shown in Fig. 73, rupture may occur along the pitch, A-D, or along two diagonal pitches and a semi-pitch, as A-B-C-D. The calculation for the value of V must be based, therefore, on an equality of strength in these two directions. The method will be given later in the deduction of Traill's formulæ.

In simple chain riveting, the minimum value of V is fixed by the same considerations which govern the minimum pitch, *i. e.*, to prevent cracking the plate and to provide room for making the rivet-point, V, minimum, should not be less than $2d$ and is preferably $2.5d$ in boiler-joints and $3d$ in structural work.

7. Margin and Lap.—To avoid cracking the plate in punching or riveting, the distance from the nearest edge of the nearest rivet-hole to the edge of a plate or strap should not be less, as

experience has shown, than the diameter of the rivet-hole. The margin is measured from the centre of the hole. The *least* value of the *margin* is therefore:

$$E = 1.5d. \tag{84}$$

The width of the *lap* depends upon the form of the joint. Thus, in Fig. 63 *a*, it is $2E$; in Fig. 63 *b*, $2E + V$; in Fig. 64 *k*, $2E + 2V_1$.

As noted previously, the rivet tends to rupture the margin, as in Fig. 70, and to shear it, as in Fig. 71. In designing the margin for rupturing stress, the portion of the plate included between the rivet and the edge may be taken, approximately, as a rectangular beam, fixed at the ends and loaded in the middle, since the rivet is slightly less in diameter than the hole and has, theoretically, but a line bearing on the walls of the latter. For such a beam, the general formulæ give:

$$M = \tfrac{1}{8} \cdot W \cdot l = S_t \cdot \frac{I}{c} \quad \text{and} \quad \frac{I}{c} = \frac{bd^2}{6},$$

in which M = maximum bending moment, W = total load, l = span, I = gravity moment of inertia of the cross-section of the beam, c = distance of most remote fibre of the cross-section from the neutral axis, b = breadth, and d = depth of beam. In the assumed beam:

Span = diameter of rivet = d;
Breadth = thickness of plate = t;
Depth = $E - d/2$ (Fig. 70);
Distance $c = \frac{1}{2}$ depth $= \frac{1}{2}(E - d/2)$;

$$\frac{I}{c} = \frac{t\left(E - \frac{d}{2}\right)^2}{6}.$$

Let F = factor of safety and $S_t \div F$ = allowable working stress. Substituting:

$$M = \frac{Wd}{8} = \frac{S_t}{F} \cdot \frac{t\left(E - \frac{d}{2}\right)^2}{6}. \tag{85}$$

It is assumed — necessarily, but practically without warrant — that the load on any pitch-section, as *m-n-o-r*, Fig. 64 *d*, is divided

equally among the rivets in that section. Thus, in the figure referred to, the total permissible load on the section is:

$$(p-d)t \cdot S_t \div F;$$

and, since there are three rivets in the section, the load per rivet is:

$$\frac{(p-d)t}{3} \cdot \frac{S_t}{F}.$$

In general, let $n =$ the number of rivets in the section. Then:

$$W = \frac{(p-d)t}{n} \cdot \frac{S_t}{F}.$$

Substituting in (85):

$$\frac{(p-d)t}{n} \cdot \frac{S_t}{F} \cdot \frac{d}{8} = \frac{S_t}{F} \cdot \frac{t\left(E-\frac{d}{2}\right)^2}{6};$$

$$E = \frac{d}{2} + \sqrt{\frac{3}{4n}(p-d)d}, \tag{86}$$

which equation applies to both lap and butt joints. In a good design, it gives a close approximation to $E = 1.5d$, as in (84).

The shearing and crushing of the portion of the plate between the edge and the rivet, as in Fig. 71, remain to be considered. For equal strength of margin with regard to these stresses, we have:

$$\textit{Resistance to shearing} = 2\left(E - \frac{d}{2}\right)t \cdot S_s;$$

$$\textit{Resistance to crushing} = d \cdot t \cdot S_c.$$

Assuming $S_s = \frac{2}{3}\, S_c$ and equating:

$$E = 1.25\, d.$$

Hence, if, as in (84), $E = 1.5d$, that width will be more than sufficient to withstand all stresses in the margin.

8. Chain and Staggered Riveting. — If the diagonal pitch be properly proportioned, there is, theoretically, no difference in strength between chain and staggered riveting, although some tests have shown a slight advantage in favor of the former. The practical advantage of staggered riveting is that pressure joints may be made tighter, owing to the reduced width of the lap. The

breadth of the latter depends upon that of the margin, E, and of the transverse pitch, V. In chain riveting, $V = 2$ to $2.5d =$ minimum pitch, p; in staggered riveting, as shown, $V = 0.4p$, theoretically, although practically it is greater.

For example, in double-riveted lap joints, Fig. 63 *b, c*, in boiler-work, with steel plates and steel rivets, with plate-thickness, $t = 0.75$ ins., rivet-diameter, $d = 1.125$ ins., and pitch, $p = 3.30$ ins.:

	Staggered.	Chain.
$E = 1.5d =$	1.69	1.69
$V =$	1.78	2.75
$2E + V = \text{Lap} =$	5.16	6.13

9. Butt-Joint, Single Strap. — This seam, Fig. 65, consists essentially of two abutting lap-joints. Theoretically, it has, in tension and shear, no more strength than the latter, and, to resist these stresses only, the strap need be no thicker than the connected plates. Its practical advantage lies in the fact that the plates are in the same plane and the bending to which they are subjected in lap-joints is largely removed. The tension on the plates will, however, tend to bend the strap; and, for this reason, the latter should be thicker than the plates, the increase depending upon the form of the riveting but being, as a minimum, ⅛ the thickness of plate, when no rivets are omitted. The sole advantage of this form of joint lies in the resistance to bending stress offered by the thick strap. Its relative cost, as compared with that of stronger joints of the double-strap type, warrants its use in exceptional cases only.

10. Butt-Joints, Double Strap. — The advantages of this joint, Fig. 64, are that, not only are the connected plates in the same plane, thus transferring bending stress from them to the straps; but the rivets, with regard to the plates, are in "double shear," *i. e.*, neither plate can withdraw from the joint without shearing, across two sections, each rivet passing through it. The efficiency of each rivet is, therefore, as compared with the lap or single butt-joint, apparently doubled in shearing, although, as will be shown later, the strength in double shear is not twice, but about 1.75 times that in single shear.

The progressive increase of efficiency of this joint with multiple riveting, is limited by the fact that the resistances of the plate and rivet to bearing stress are not doubled with that to shearing. For

example, consider a $1\frac{1}{8}$-in. rivet passing through $\frac{3}{4}$-in. steel plate. Take $S_s = 44,000$ and $S_c = 70,000$. Then:

	Single Shear.	Double Shear.
Ultimate bearing load $= d \cdot t \cdot S_c =$	59,080	59,080
" shearing " $= \pi d^2/4 \cdot S_s =$	43,736	
" " " $= \pi d^2/4 \cdot S_s \times 1.75 =$		76,538

In single shear, the rivet has an ultimate strength of 43,736 lbs.; in double shear, of 76,538 lbs. The bearing strength is the same in both cases, leaving, in double shear, a surplus shearing strength of 17,458 lbs., which is unavailable, since the limit in bearing has been reached. The data, as above, will be regarded simply as an indication of the principle involved.

The double-strap joint has also, and in greater degree, the advantage of the single-strap joint in relieving the plates of bending stress. With regard to tension and shear only, the thickness of the straps need be but one half that of each connected plate. Owing to bending stress, however, as in the single-strap joint, each strap should have, as a minimum, $\frac{5}{8}$ the thickness of the plate, when no rivets are omitted.

11. Butt-Joint, Unequal Straps.—The butt-joint with double straps unequal in width, Fig. 61, is stronger than a joint with equal straps of the narrower width, owing to the added rivets of wider pitch in the outer row. A properly designed joint of any type is so proportioned as to take full advantage of the tensile strength of the net section of metal along the pitch section. In double shear, the shearing resistance, as shown, grows more rapidly than the tensile and bearing resistances of the joint. Hence, by increasing the width of the inner strap and adding two rivets, doubly spaced, as at A and B, Fig. 61, the length of the net plate-section becomes $p - d$, instead of $p/2 - d$, as at C-D, and the bearing resistance is increased also, without adding more shearing strength than that given by one rivet in single shear. The main purpose of this type of joint is to enlarge the net plate-section, as at A-B.

This joint has met wide adoption in stationary and locomotive boilers. It has a practical advantage in that, as the wider strap is always placed inside, there is a section of metal within the shell-sheet and back of the calking edge on the outer strap. If, through bad calking, the shell-sheet be indented on the outside, the inner strap acts as a support instead of providing an edge over which

the shell-sheet may bend and crack. The half-pitch, at *C-D*, ensures tightness.

12. LAP-JOINT, SINGLE STRAP. — This joint, Fig. 66, consists usually of an ordinary single-riveted lap-joint with, on the inside, a butt-strap or welt-strip covering its whole length. There are three rows of rivets. Those in the centre pass through both plates and the strap; those in one outer row extend through the strap and one plate; and those in the other row, through the strap and the other plate. The pitch of the outer rows is twice that in the centre.

This joint is intermediate between the lap and butt types. Its narrow central pitch ensures tightness while the wide pitch of the outer rows gives increased length of net plate-section. The inner edge of the main seam is inaccessible; but the joint is stronger than the lap-form in tension and shearing and the strap makes it stiffer against bending. The strap, when below the water-line in a boiler, prevents in a lap-joint the action known as "furrowing," *i. e.*, the corrosion in grooves of the plate-metal near the joint.

13. GROUP RIVETING, Fig. 67. — This form of joint is applicable especially in structural work for splicing narrow plates, etc. The stresses in the net section of plate decrease from Row No. II. onward and, in a lap-joint, efficiencies varying with the number of rows and ranging from 80 per cent. upward are obtained. The rivets are disposed in groups according to an arithmetical series, the number in the rows being:

1, 2, 1;
or, 1, 2, 3, 2, 1;
or, 1, 2, 3, 4, 3, 2, 1, etc.

42. The Theoretical Strength of Riveted Joints.

The riveted joint is a structure whose character and methods of manufacture forbid extreme refinement of design. The plates are not only exposed in various parts to direct tension, compression, and shear, and the rivets to the latter two stresses in addition to their initial tension; but there is also, in service, bending action on the rivets and on the plates or straps. Even on the assumption of perfect workmanship throughout, the relation of the various stresses, with regard to any element of the joint, is so complex that a fair approach to the value of the resultant stress could be obtained only by intricate calculation. Again, assuming such a

calculation as possible in practical design, its results would be affected materially by the process of manufacture of the joint, which process is essentially of such a nature as to exclude the accurate fitting and alignment which are required for the correct distribution of the total load among the rivets, plates, and straps.

The many tests of joints give information of much value. That information is, however, conclusive only in revealing the *apparent* stress at which certain elements of that particular joint yielded. The *actual* stress which existed within those elements at destruction is, owing to hidden components, unknown ; and the rearrangement of stresses which occurs at various stages of a test renders impossible an accurate determination of anything more than the apparent load on any element at any time.

Furthermore, the majority of published tests have been made upon thin plates which develop, as a rule, maximum ultimate resistance ; the bulk of the test specimens have been narrow sections of the joint ; and the method of testing is direct tension on a plane specimen. In practice, on the other hand, the plate is, in boiler-work, if thin, of small diameter and great curvature ; or, for high pressures, may be of large diameter and less curvature but of maximum thickness. Again, in structures, a joint — for example, that in the web of a plate-girder — may be in tension at one end and in compression at the other ; or, as in the multiple plate form of chord construction, it may present conditions which differ widely from those of the simple joint tested in tension.

While, therefore, actual experiments upon joints give the only available knowledge of their final strength, the use in designing of the results thus obtained should be governed by the conditions of the specimen tested and of the joint required. As a rule, practical designing regards only the simple stresses in plates and rivets ; neglects, except in the added thickness of butt-straps, the bending action within the joint ; allows for these omitted stresses by the use of a fair factor of safety ; and accepts the efficiency of the joint as thus computed.

1. NOTATION. — In the discussion of joint-strengths which follows :

d = diameter of rivet ;

n = number of rivets in the pitch-section, *i. e.*, a strip of joint of length p ;

t = thickness of plate;
t_1 = thickness of single butt-strap = $1\frac{1}{8}t$;
t_2 = thickness of double butt-strap = $\frac{5}{8}t$;
p = pitch, longitudinal, greatest;
p_d = pitch, diagonal;
V = pitch, transverse, in staggered riveting, with no rivets omitted, and in chain riveting, when alternate rivets are omitted in outer row;
V_1 = distance between outer and next row of rivets in staggered riveting, when alternate rivets are omitted in outer row;
E = width of margin = $1.5d$;
c = shearing constant = 1 for lap and single butt-strap joints and 1.75 for double butt-strap joints;
R = ultimate resistance in tension of a strip of solid plate, the width of the greatest pitch, p;
R_t = ultimate resistance, in tension, of joint;
R_s = ultimate resistance, in shearing, of joint;
R_c = ultimate resistance, in bearing, of joint;
R_m = ultimate resistance of joint, with alternate rivets in outer row omitted, when joint yields by tearing the plate along the central row and shearing one rivet in the outer row;
S_t = ultimate unit tensile stress of plate;
S_s = ultimate unit shearing stress of rivet;
S_c = ultimate unit bearing stress of rivet and plate;
E_t = tensile efficiency, per cent., of joint = $R_t/R \times 100$;
E_s = shearing efficiency, per cent., of joint = $R_s/R \times 100$;
E_c = bearing efficiency, per cent., of joint = $R_c/R \times 100$;
E_m = efficiency, per cent., corresponding with the ultimate resistance, R_m, or $E_m = R_m/R \times 100$.

2. Lap-Joint, Single Riveted. — Fig. 63 *a*. In this joint, $n = 1$. We have:

$$R_t = (p - d)t \cdot S_t;$$

$$R_s = \frac{\pi d^2}{4} \cdot S_s;$$

$$R_c = d \cdot t \cdot S_c;$$

$$R = p \cdot t \cdot S_t.$$

For equal strength throughout:

$$R_t = R_s = R_c, \text{ i. e.,}$$

$$(p-d)t \cdot S_t = d \cdot t \cdot S_c; \tag{87}$$

$$\frac{\pi d^2}{4} \cdot S_s = d \cdot t \cdot S_c. \tag{88}$$

From (87):

$$p = \frac{S_c}{S_t} d + d. \tag{89}$$

From (88):

$$d = \frac{S_c}{S_s} \cdot \frac{t}{0.7854}. \tag{90}$$

$$E_t = \frac{R_t}{R} \times 100 = \left(\frac{p-d}{p}\right) 100;$$

$$E_s = \frac{R_s}{R} \times 100 = \left(\frac{S_s}{S_t} \cdot \frac{.7854 d^2}{pt}\right) 100;$$

$$E_c = \frac{R_c}{R} \times 100 = \left(\frac{S_c}{S_t} \cdot \frac{d}{p}\right) 100.$$

It will be noted that (89) and (90) are derived by equating the bearing strength of the joint to its tensile and shearing strengths. As will be shown later, the ultimate compressive or bearing strength of a rivet in its hole or of the walls of the hole itself, is a somewhat uncertain quantity, since the confined situation of the metals restricts their plastic flow. The exact manner in which the "bearing pressure" acts upon the surface of the rivet or of the hole is unknown. It is assumed, with some warrant, to be equivalent to a total pressure uniformly distributed over the projected semi-intrados, or $d \times t$. Owing to the uncertainty in this matter, some designers assume, from experience, a ratio d/t and find p by equating R_t and R_s, thus:

$$R_t = R_s = (p-d)t \cdot S_t = 0.7854\, d^2 \cdot S_s;$$

$$\frac{p}{d} = 1 + .7854 \frac{d}{t} \cdot \frac{S_s}{S_t}. \tag{91}$$

In this equation, t is known from prior considerations; d/t is assumed, giving the value of d; S_s/S_t is ascertained from tests of the metals. The value of p can be found, therefore, by substitution.

Example: Steel plate and rivets; $t = \frac{3}{8}$ in.; $S_t = 65{,}000$; $S_s = 0.8S_t = 52{,}000$; $S_c = 70{,}000$.

From (90):

$$d = \frac{70}{52} \times \frac{0.375}{0.7854} = 0.643, \text{ say } \frac{21}{32} \text{ in.} = 0.656.$$

From (89):

$$p = \frac{70}{65} \times 0.656 + 0.656 = 1.363, \text{ say } 1\tfrac{3}{8} \text{ in.}$$

Substituting the values of d and p in the equations for efficiencies:

$$E_t = 52.27 \text{ per cent.};$$
$$E_s = 52.44 \text{ per cent.};$$
$$E_c = 51.39 \text{ per cent.}$$

Efficiency of Joint = 51.39 per cent., the least of the three efficiencies, as above.

3. LAP JOINT, DOUBLE RIVETED, NO RIVETS OMITTED. — Fig. 63, *b*, *c*. In this joint, $n = 2$. We have:

$$R_t = (p - d)t \cdot S_t;$$
$$R_s = (.7854\ d^2 \cdot S_s)n;$$
$$R_c = (d \cdot t \cdot S_c)n;$$
$$R = p \cdot t \cdot S_t.$$

For equal strength throughout:

$$R_t = R_s = R_c$$

$$\therefore (p - d)t \cdot S_t = n \cdot dtS_c; \tag{92}$$

$$n \times 0.7854d^2S_s = n \cdot dtS_c. \tag{93}$$

From (92):

$$p = \frac{S_c}{S_t} \cdot nd + d. \tag{94}$$

Equation (93) gives equation (90) as before.

$$E_t = \frac{R_t}{R} \times 100 = \left(\frac{p - d}{p}\right) 100, \tag{95}$$

$$E_s = \frac{R_s}{R} \times 100 = n\left(\frac{S_s}{S_t} \cdot \frac{0.7854d^2}{pt}\right) 100, \tag{96}$$

$$= \frac{R_c}{R} \times 100 = n\left(\frac{S_c}{S_t} \cdot \frac{d}{p}\right) 100. \tag{97}$$

Example.—Double-riveted seam, chain or staggered; steel plate and rivets; $t = \frac{7}{16}$ in.; S_t, S_s, S_c are, respectively, 65,000, 52,000, 70,000.

From (90):

$$d = \frac{70}{52} \times \frac{0.4375}{0.7854} = 0.75 = \tfrac{3}{4} \text{ in.}$$

From (94):

$$p = \frac{70}{65} \times 2 \times 0.75 + 0.75 = 2.366, \text{ say } 2\tfrac{3}{8} \text{ in.}$$

Substituting in (95), (96), (97):

$$E_t = 68.42 \text{ per cent.};$$
$$E_s = 68.01 \text{ per cent.};$$
$$E_c = 68.02 \text{ per cent.}$$

Efficiency of Joint = 68.01 per cent., a gain of 68.01 − 51.39 = 16.62 per cent. over the single-riveted seam.

It will be observed that equations (90), (94), (95), (96), (97) are general.

The double-riveted seam, with alternate rivets omitted in the outer row, is not used with lap-joints.

4. Lap Joint, Treble Riveted, no Rivets Omitted, chain or staggered, as in Fig. 63 *d, e*. In this joint, $n = 3$.

Example.—Steel plate and rivets; $t = \frac{1}{2}$ in.; S_t, S_s, S_c are, respectively, 65,000, 52,000, 70,000.

From (90):

$$d = \frac{70}{52} \times \frac{0.5}{0.7854} = 0.856, \text{ say } \tfrac{13}{16} \text{ in.}$$

From (94):

$$p = \frac{70}{65} \times 3 \times \frac{13}{16} + \frac{13}{16} = 3.432, \text{ say } 3\tfrac{7}{16} \text{ in.}$$

From (95), (96), (97):

$$E_t = 76.36 \text{ per cent.};$$
$$E_s = 72.22 \text{ per cent.};$$
$$E_c = 76.25 \text{ per cent.}$$

Efficiency of Joint = 72.22 per cent., a gain of 72.22 − 68.01 = 4.21 per cent. over the double-riveted seam. Compare with

5. Lap Joint, Treble Riveted, Alternate Rivets Omitted in Outer Rows. — In this case, the grouping of the rivets is that shown, on each side of the seam, in Fig. 64, *h* or *k*. In this joint, $n = 4$.

Example. — Data, the same as in the previous case. Since (90) does not include n, we have, as before, $d = \frac{13}{16}$ in. From (94):

$$p = \frac{70}{65} \times 4 \times \frac{13}{16} + \frac{13}{16} = 4.31, \text{ say } 4\tfrac{5}{16} \text{ in.}$$

From (95), (96), (97):

$$E_t = 81.16 \text{ per cent.};$$
$$E_s = 76.74 \text{ per cent.};$$
$$E_c = 81.42 \text{ per cent.}$$

With regard to this form of riveting, a further efficiency must be considered. Since there are twice as many rivets on the central as on either outer row, the net plate-section on the central row is less and the plate might tear along that line. Before total yielding can occur in this manner, however, one rivet in the outer row must be sheared for each pitch-section. Hence, with regard to rupture in this manner, the total resistance, R_m, of the joint is equal to that in tension of the net section along the middle row, plus that in shearing of one rivet in the outer row. We have for these conditions:

$$R_t \text{ (central row)} = (p - 2d)t \cdot S_t;$$
$$R_s \text{ (one rivet, outer row)} = 0.7854d^2 \cdot S_s;$$
$$R_m = R_t + R_s = (p - 2d)t \cdot S_t + 0.7854d^2 \cdot S_s. \qquad (98)$$

Similarly to (95), etc., the efficiency under these conditions is:

$$E_m = \frac{R_m}{R} \times 100 = \left[\frac{p - 2d}{p} + \frac{S_s}{S_t} \cdot \frac{0.7854d^2}{pt}\right] 100. \qquad (99)$$

Substituting in (99):

$$E_m = 81.5 \text{ per cent.}$$

Comparing this with the efficiencies deduced previously, *Efficiency of Joint = 76.74 per cent.*, a gain of $76.74 - 72.22 = 4.52$ per cent., by omitting alternate rivets in the outer rows.

The elements of the two joints are with:

	t	d	n	p (*middle*)	p (*outer*)
No rivets omitted	$\frac{1}{2}$	$\frac{13}{16}$	3	$3\frac{7}{16}$	$3\frac{7}{16}$
Alt. rivets omitted	$\frac{1}{2}$	$\frac{13}{16}$	4	$2\frac{5}{32}$	$4\frac{5}{16}$

The plate-thickness and rivet-diameter are the same in both cases. In the modified joint, one rivet has been added and the pitch of the outer rows made greater and that of the central row less. The addition of the rivet increases the strength in shearing and bearing, since the outer pitch has not been increased proportionately. The extended outer pitch augments the net plate-section and, hence, the tensile strength. The reduced pitch and tensile strength of the central row are more than balanced by the added resistance of one rivet in shearing before rupture can occur along that row. Hence, the modified joint is, in every respect, the stronger. Its only disadvantage lies in the lessened tightness, due to wider pitch, on the outer rows, to offset which there is a reduced pitch in the centre.

6. Butt-Joint, Single Strap, Fig. 65. — This joint is, in effect, composed of two abutting lap-joints, one between the strap and each plate. Hence, with regard to the plates, the methods of design used for lap-joints, apply. This is true also of the strap, since the increased thickness given it is intended solely to resist bending stress, which stress analysis does not consider. Under these conditions, there is, theoretically, no gain in efficiency over the lap-joint, except that the bearing and tensile stresses in the strap are less. The increased strength against bending action is more than offset by the additional cost of the strap and of calking one more seam. This joint, therefore, is seldom used.

7. Butt-Joints, Double Straps of Equal Width, Fig. 64.— This joint is the strongest form for any purpose. The straps, one on each side of the plates, reduce the bending action to a minimum and the rivets are in double shear. The single-riveted type, Fig. 64 *a*, is seldom used for boiler work, since, with but one row of rivets, the latter must be, for strength, of relatively large diameter and therefore so widely spaced as to interfere with tightness. The efficiency of such a joint is only about 65 per cent., or less than that of the double-riveted lap form, with usual proportions in both cases.

The butt-joint with double straps consists essentially of two double shear joints with regard to the plates and of four single shear lap-joints with regard to the straps. Each strap is assumed to bear one half of the total load upon the plates and is, as stated previously, made $\frac{5}{8}$ the thickness of the plate, as a minimum, when no rivets are omitted, in order to provide for bending stress. The relations between the loads and strengths of the plate and either strap are:

		Plate.	Strap.
Load,		1	$\frac{1}{2}$
Strength,	tensile,	1	$\frac{5}{8}$
"	shearing,	1.75	1
"	bearing,	1	$\frac{5}{8}$

With one half the load of the plate, the strap has five eighths the strength of the latter in tension and bearing and its shearing strength exceeds its load in the proportion of 1 to 0.875. Hence, if the joint be designed properly for the plates, its strength will be ample for the straps.

The general formulæ deduced previously for lap-joints were founded solely upon the size and grouping of the rivets with regard to the seam. Precisely the same conditions hold in butt-joints, double-strapped, with the additional consideration that the factor, $c = 1.75$, is introduced, since, with regard to the plates, the rivets are in double shear. Therefore, for the plates of such joints, we have:

$$R_t = (p - d)\,t \cdot S_t;$$
$$R_c = (d \cdot t \cdot S_c)n;$$
$$R_s = (0.7854\, d^2 \cdot S_s \times 1.75)n;$$
$$R = p \cdot t \cdot S_t.$$
$$R_t = R_s = R_c$$
$$\therefore (p - d)t \cdot S_t = ndt \cdot S_c;$$
$$1.374nd^2 \cdot S_s = ndt \cdot S_c;$$

whence:

$$p = \frac{S_c}{S_t} \cdot nd + d; \tag{94}$$

$$d = \frac{S_c}{S_s} \cdot \frac{t}{1.374}; \tag{100}$$

$$E_t = \frac{R_t}{R} \times 100 = \left(\frac{p-d}{p}\right)100; \qquad (95)$$

$$E_s = \frac{R_s}{R} \times 100 = n\left(\frac{S_s}{S_t} \cdot \frac{1.374d^2}{pt}\right)100. \qquad (101)$$

$$E_c = \frac{R_c}{R} \times 100 = n\left(\frac{S_c}{S_t} \cdot \frac{d}{p}\right)100. \qquad (97)$$

Where alternate rivets are omitted in the outer row, the plate may tear along the central row; but, before yielding occurs, one rivet, in double shear, must be sheared in the outer row. We have for these conditions:

$$R_t(\text{central row}) = (p - 2d)t \cdot S_t;$$

$$R_s(\text{one rivet, outer row}) = 1.374d^2 \cdot S_s;$$

$$R_m = R_t + R_s = (p-2d)t \cdot S_t + 1.374d^2 \cdot S_s;$$

$$E_m = \frac{R_m}{R} \times 100 = \left[\frac{p-2d}{p} + \frac{S_s}{S_t} \cdot \frac{1.374d^2}{pt}\right]100. \qquad (102)$$

From the equations given above — if the thickness of the plate and physical characteristics of the metal be known — the diameter and pitch of the rivets and the various efficiencies of any required butt-joint with double straps, may be obtained. When alternate rivets in the outer row are omitted, the butt-strap must be thicker than when no rivets are omitted, in the ratio $(p - d) \div (p - 2d)$, as will be shown later in the deduction of Traill's formulæ.

The efficiencies of butt-joints, double-strapped, with steel rivets and plates, as computed by Traill * for cylindrical boilers, with various thicknesses of plates and diameters and pitches of rivets, are:

Single-riveted,		59.35 to 65.00 per cent.
Double "		73.39 " 78.00 " "
" "	each alternate rivet in outer row omitted	79.68 " 82.32 " "
Treble "		79.68 " 82.32 " "
" "	each alternate rivet in both outer rows omitted	83.24 " 84.66 " "
" "	each alternate rivet in the outer row omitted	84.95 " "
Quadruple "		83.24 " 84.66 " "

* "Boilers: Marine and Land," 1896, pp. 306–318.

Example.—U. S. Cruiser *Raleigh;* diameter of cylindrical boilers, 14 ft., 6 ins.; steam-pressure, 160 lbs., gauge; longitudinal seam, butt-joint, double-strapped, treble riveted, alternate rivets in outer rows omitted (Fig. 64 *k*); $S_t = 65{,}000$; $S_c = 70{,}000$; $S_s = 44{,}000$; factor of safety $= 4.5$. From the considerations as to the shell (§ 44), the plate-thickness, $t = 1\frac{3}{16}$ in. In this joint, $n = 4$.

From (100):

$$d = \frac{70}{44} \cdot \frac{1.187}{1.374} = 1.375 = 1\tfrac{3}{8} \text{ins.}$$

From (94):

$$p = \tfrac{70}{65} \times 4 \times \tfrac{11}{8} + \tfrac{11}{8} = 7.298 = \text{say, } 7\tfrac{5}{16} \text{ in.}$$

The corresponding particulars of this joint, as built, were:

$$t = 1\tfrac{15}{64} \text{ in.}; \; d = 1\tfrac{5}{16} \text{ in.}; \; p = 7\tfrac{1}{4} \text{ in.},$$

an allowance of $\frac{3}{64}$ in. for corrosion having been made on the calculated thickness, t.

8. Butt-Joint, Double Straps of Unequal Widths, Fig. 61. This joint, as designed for locomotive boilers, is shown in Fig. 76.

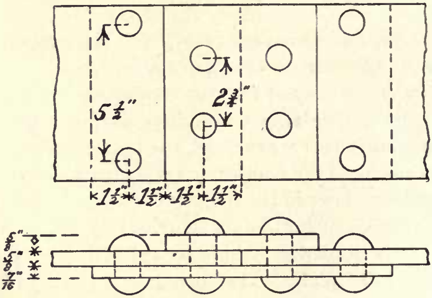


Fig. 76.

The rivets are 1 in. in diameter and of wrought iron. The plates and straps are of steel. The thickness of plates and of outer strap is $\frac{5}{8}$ in.; of inner strap, $\frac{7}{16}$ in. Take $S_t = 60{,}000$ and $S_s = 48{,}000$. Disregard resistance to crushing. In this joint, $n = 3$. Failure may occur by:

(*a*) Rupture of plate along outer row of rivets. (*b*) Shearing one rivet, outer row (single shear) and two rivets, inner row

(double shear). (c) Rupture of plate along inner row and shearing one rivet, outer row (single shear).

For these conditions:

$$R_t = (p - d)t \cdot S_t;$$
$$R_s = 0.7854 d^2(1 + 2 \times 1.75)S_s;$$
$$R_m = (p - 2d)t \cdot S_t + 0.7854 d^2 \cdot S_s;$$
$$R = p \cdot t \cdot S_t.$$

Equating R_t and R_m: $d = 0.995$, say 1 in.
Equating R_t and R_s: $p = 5.52$, say $5\frac{1}{2}$ in.
The efficiencies are:

$$E_t = \frac{R_t}{R} \times 100 = 81.82 \text{ per cent.};$$

$$E_s = \frac{R_s}{R} \times 100 = 82.25 \text{ per cent.};$$

$$E_m = \frac{R_m}{R} \times 100 = 81.92 \text{ per cent.}$$

With regard to crushing, failure may occur by:

(d) Rupture of plate along inner row and crushing one rivet in outer row and inner strap; (e) crushing two rivets in inner row in plate or outer strap and one rivet in outer row and inner strap.

The total resistances of the joint are:

For conditions (d):

$$(p - 2d)t \cdot S_t + (1 \times \tfrac{7}{16})S_c.$$

For conditions (e):

$$2(1 \times \tfrac{5}{8})S_c + (1 \times \tfrac{7}{16})S_c.$$

Investigation of these resistances shows that S_c must have a value of about 100,000 lbs. per sq. in. in order to make the efficiency of the joint about equal to those already deduced. The moderate value (70,000) used previously with steel rivets will not serve, therefore, in this case. Since the destructive limit of metal wholly free to flow, differs from that of the same metal when plastic flow is restricted, as in a riveted joint, considerable variations in the value of S_c may be expected. Professor Kennedy recommends 96,000 lbs. per sq. in., and Mr. Stoney 50 tons per sq. in., for the plates in butt-joints, double-strapped.

Failure of the lap-seams of the straps of this joint may occur (disregarding crushing stress), as follows:

Inner Strap:

(f) By shearing 3 rivets in single shear;

(g) By shearing 1 rivet in single shear and tearing the plate along the inner row of rivets.

Outer Strap:

(h) By shearing 2 rivets in single shear;

(k) By tearing the plate along the inner row of rivets.

From (f) and (h) it is clear that, in shearing, the seam of the inner strap is necessarily the stronger in this form of joint. With straps of equal thickness, this seam is also the stronger with regard to (g) and (k). Hence, for equality in resistance to these two methods of failure, the thickness of the inner strap may be made less than that of the outer, as in Fig. 76.

The purpose, in extending the inner strap so that another row of rivets in double pitch may be added, is that the net plate-section — which is then taken along the added row — may be increased, thus giving greater tensile efficiency. The latter, with this type, is relatively high. Thus, the joint shown in Fig. 76 is, as compared with butt-joints having straps of equal width, stronger than the double-riveted and but little weaker than the treble-riveted forms, no rivets being omitted in either of the latter cases.

9. Single-strapped Lap Joint, Fig. 66.—The analysis of this joint is given briefly in Reuleaux's *Constructor.** The joint consists of an ordinary single-riveted lap-seam with an inside strap riveted to the shell plates at, and on each side of, the seam. The pitch of the outer rows is twice that of the central line of rivets. The central rivets pass through the strap.

Against stress normal to the seam, three rivets act — two in the lap-joint and one passing through the strap. The assumption is that the total load, P, upon the pitch-section is divided equally among the three rivets and that, hence, the strap withstands one third of that load and the joint the remaining two thirds. Therefore, if the strap and plates be of equal thickness and S_t be the ultimate unit tensile stress in the plate beyond the joint, the corresponding stress in the *unperforated* plate, within the joint and between the rivet-rows, will be $\frac{2}{3}S_t$ and, in the strap, $\frac{1}{3}S_t$. Again, although the rivets at the lap-seam pass through the strap, the latter is, at

* Suplee's translation, 1895, p. 43.

that seam, pulled in opposite directions by equal forces and hence transmits no stress to the central row of rivets.

Assuming equal thicknesses and tensile strengths throughout, the ultimate unit resistance of the plate, within and without the joint, and of the strap would be equal under similar conditions; but, in this type of joint, the conditions are such that the pitch-section beyond the joint bears the total load, P, the unperforated plate within the joint bears $\frac{2}{3}P$, and the strap $\frac{1}{3}P$. Hence, with equal thickness, the plate within the joint and the strap — as compared with the plate beyond the joint — are equivalent, respectively, to metal, $1 \div \frac{2}{3} = \frac{3}{2}$ and $1 \div \frac{1}{3} = 3$ times as strong. We have, then, for the ultimate tensile resistances and efficiencies:

$$R \text{ (plate beyond joint)} = p \cdot t \cdot S_t;$$

$$R_t \text{ (plate within joint)} = \tfrac{3}{2}(p - 2d)tS_t;$$

$$R_t \text{ (strap)} = 3(p - 2d)tS_t.$$

$$E_t \text{ (plate)} = \frac{R_t \text{(plate)}}{R} \cdot 100 = \tfrac{3}{2}\frac{(p - 2d)}{p} 100;$$

$$E_t \text{ (strap)} = \frac{R_t \text{(strap)}}{R} \cdot 100 = 3\frac{(p - 2d)}{p} 100.$$

The efficiency of the strap-metal will depend upon its resistance on the central row, along which line it will tend to part.

In shearing stress, in order that the right hand plate may withdraw from the joint, it must shear one strap and two lap rivets, all in single shear. For the strap the conditions are the same. Hence, taking $S_s = 0.8S_t$:

$$R_s = 3 \cdot \frac{\pi d^2}{4} \cdot S_s;$$

$$E_s = \frac{R_s}{R} \cdot 100 = \tfrac{3}{5} \cdot \frac{\pi d^2}{pt} \cdot 100.$$

Comparing the efficiencies with those of an ordinary double-riveted lap-joint, as in (95), (96):

	E_t	E_s
Single-strapped lap-joint:	$\frac{3}{2} \cdot \frac{p - 2d}{p} \cdot 100;$	$\frac{3}{5} \cdot \frac{\pi d^2}{pt} \cdot 100;$
Double-riveted lap-joint:	$\frac{p - d}{p} \cdot 100;$	$\frac{2}{5} \cdot \frac{\pi d^2}{pt} \cdot 100.$

10. Group Riveting, as shown in the lap-joint, Fig. 67, is an arrangement of the rivets in an arithmetical series, which grouping — on the assumption of an equal division of the load among the rivets — gives a gradually reducing tensile stress in the plate from one side to the other of the group. The substance of the following analysis is taken from Reuleaux's *Constructor.**

Referring to Fig. 67, let P be the total load on the pitch-section, p; m, the number of rivets in the central row; and a, the pitch of that row. Then $p = m \times a$. From the properties of an arithmetical series, the total number of rivets in the group $= m^2$. In Fig. 67, $m = 4$.

(*a*) *Shearing and Greatest Tensile Stresses.* — To find the ratio between pitch, p, diameter, d, and thickness, t, which shall give equality between the tensile strength of the net plate-section in the first, or outer, row and the shearing strength of the joint, let S_t be the unit tensile stress in the plate beyond the joint and S_t^I that in the outer row. Then:

$$R_t \text{ (outer row)} = (ma - d)t \cdot S_t^I;$$

$$R_s \text{ (joint)} = m^2 \cdot \frac{\pi d^2}{4} \cdot S_s.$$

Equating and taking $S_s = 0.8 S_t^I$:

$$m^2 \pi d^2 = 5(ma - d)t$$

$$\left.\begin{aligned} \frac{a}{t} &= m \cdot \frac{\pi}{5}\left(\frac{d}{t}\right)^2 + \frac{1}{m} \cdot \frac{d}{t} \\ \frac{a}{d} &= m \cdot \frac{\pi}{5}\left(\frac{d}{t}\right) + \frac{1}{m} \end{aligned}\right\} \qquad (50,\ C)$$

From which, $p = ma$ may be found.

(*b*) *Tensile Stress in any Row.* — To find the stress on the net plate-section in any given row, there must be deducted — on the assumption of an equal division of the load among the rivets — the fraction of the total load, P, which is borne by preceding rivets. Thus, in Fig. 67, the net plate-section of Row I. carries the full load, P; that of Row II., P minus the load, $\frac{P}{m^2}$, on the single rivet

* Suplee's translation, 1895, p. 41.

of Row I.; that of Row III., P minus loads on Rows I. and II., or $\frac{3P}{m^2}$, etc.

The fractions of the total loads borne by the net plate-sections of the various rows will then be:

$$\text{I., } \frac{m^2}{m^2}\cdot P;\ \text{II., } \frac{m^2-1}{m^2}\cdot P;\ \text{III., } \frac{m^2-3}{m^2}\cdot P;\ \text{IV., } \frac{m^2-6}{m^2}\cdot P, \text{ etc.}$$

Let the unit stresses in the net plate-sections, beginning with the outer, be S_t^I, S_t^{II}, S_t^{III}, S_t^{IV}, etc. Then, equating the loads and resistances:

$$\left.\begin{array}{ll} \text{Row I.:} & P = (ma - d)t \cdot S_t^I; \\ \text{Row II.:} & \frac{m^2-1}{m^2}\cdot P = (ma - 2d)t \cdot S_t^{II}; \\ \text{Row III.:} & \frac{m^2-3}{m^2}\cdot P = (ma - 3d)t \cdot S_t^{III}; \\ \text{Row IV.:} & \frac{m^2-6}{m^2}\cdot P = (ma - 4d)t \cdot S_t^{IV}; \text{ etc.} \end{array}\right\} \quad (51, \text{C})$$

Now, let a bear such a proportion to d that $S_t^I = S_t^{II}$. To find the value of this ratio in terms of m, equate the values of P from the first and second equations, as above:

$$(ma - d)t = (ma - 2d)t \cdot \frac{m^2}{m^2 - 1}$$

$$\frac{a}{d} = \frac{m^2 + 1}{m}, \quad (52, \text{C})$$

which equation holds only for the condition that $S_t^I = S_t^{II}$. Equating the value of P for each succeeding row with that for the first row and substituting the value of $\frac{a}{d}$:

$$\frac{S_t^{III}}{S_t^I} = \frac{m^2-3}{m^2-2};\ \frac{S_t^{IV}}{S_t^I} = \frac{m^2-6}{m^2-3};\ \frac{S_t^V}{S_t^I} = \frac{m^2-10}{m^2-4};\ \text{etc.}$$

With $m = 4$, these ratios are, respectively, $\frac{13}{14}$, $\frac{10}{13}$, and $\frac{6}{12}$, showing that, from the second row inward, there is a gradually decreasing stress in the plate.

In order to compare the results for any given ratio between diameter and thickness with varying values of m, assume in (50, C) any convenient value for this ratio, as:

$$\frac{d}{t} = \frac{5}{\pi} = 1.5916 = \text{say } 1.6 \qquad (53, C)$$

Also, since S_t is the unit tensile stress in the unperforated plate and S_t^I the greatest unit tensile stress in any net plate-section of the joint:

$$E_t\,(\text{joint}) = \frac{S_t^I}{S_t} = \frac{ma - d}{ma} = 1 - \frac{1}{m} \cdot \frac{d}{a} = \frac{m^2}{m^2 + 1}. \qquad (54, C)$$

Again, for the unit bearing stress, S_b, we have:

$$P = m^2\,(dt \times S_b);$$

$$S_b = \frac{P}{m^2 dt}. \qquad (55, C)$$

Substituting various values of m in equations (50, 52, 53, 54, C):

$m =$	2.	3.	4.	5.
$d/t =$	1.6	1.6	1.6	1.6
$a/d =$	2.50	3.33	4.25	5.20
$a/t =$	4.00	5.32	6.80	8.32
$E_t =$	0.80	0.90	0.94	0.96

These efficiencies are based upon the assumptions that the load is divided equally among the rivets; that a bears such a proportion to d that $S_t^I = S_t^{II}$; that $S_s = 0.8 S_t^I$; and that $d/t = 1.6$.

In butt-joints, with single or double straps, there will be two groups, one on each side of the seam. Such a joint has, practically, a further advantage in the increased thickness of the strap.

43. General Formulæ for Boiler-Joints.

The formulæ deduced in this section are given in *Boilers: Marine and Land*, by Thomas W. Traill, F.E.R.N., in Foley's *Mechanical Engineers' Reference Book*, and in Seaton's *Manual of Marine Engineering* (1890). In the work referred to, Traill gives a complete series of tables calculated from these formulæ, from which the

proportions of a joint for any given boiler-diameter and steam-pressure can be selected at once without preliminary computation. These tables are of much value and have met extensive use by designers in the United States and Great Britain. No deduction of the formulæ is given in the works noted above. That which follows is, in substance, that prepared by Lieutenant Commander F. J. Schell,* U. S. Navy, for the use of midshipmen at the U. S. Naval Academy.

The notation used by Traill is:

p = pitch (greatest) of rivets in inches;
d = diameter of rivets in inches;
c = a shearing constant whose value is 1 for lap or single butt-strap, and 1.75 for double butt-strap joints;
A = area in sq. ins. of cross-section of one rivet;
n = number of rivets in section of length, p;
$\%$ = percentage of plate left between rivets in greatest pitch;
$\%_1$ = percentage of rivet-section as compared with solid plate;
$\%_2$ = percentage of combined plate and rivet-section when alternate rivets are omitted in outer row;
p_d = diagonal pitch;
E = distance from centre of nearest row of rivets to edge of plate;
V = transverse distance between rows of rivets in ordinary zigzag (staggered) riveting and in chain riveting with alternate rivets omitted in outer row;
V_1 = distance between outer and next row of rivets in zigzag riveting with alternate rivets omitted in outer row;
T = thickness of plate in inches;
T_1 = thickness of single butt-strap in inches;
T_2 = thickness of double butt-strap in inches;

1. Assumptions.—(*a*) That the mean tensile strength of steel plate is 28 tons per square inch of net section; (*b*) that the mean shearing strength of steel rivets is 23 tons per square inch of cross-section; (*c*) that a rivet in double shear offers 1.75 times the resistance to shearing opposed by a rivet in single shear; (*d*) the bearing stress is not considered.

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* *Jour. Am. Soc. Naval Engineers*, IV., 403.

2. PERCENTAGE STRENGTH OF JOINT. — The net plate-section along the greatest pitch is $(p-d)T$; the sectional area of the same length of solid plate is $p \times T$. Hence:

$$\% = 100 \cdot \frac{(p-d)T}{pT} = 100 \frac{p-d}{p}. \tag{103}$$

The resistance to shearing offered by the rivets in one pitch section is $23 \times A \times n \times c$; the tensile strength of the solid plate of length p is $28 \times p \times T$. Hence:

$$\%_1 = 100 \cdot \frac{23A \cdot n \cdot c}{28p \cdot T}. \tag{104}$$

Consider now the case in which alternate rivets are omitted in the outer row, as in Fig. 73. In the next row, the net plate-section will be $(p-2d)T$ and $100 \cdot (p-2d)/p$ will be the percentage strength of this row as compared with the solid plate. Suppose, however, that the plate tears along this row, as at K. Then, before total failure of the joint occurs, one rivet, for each pitch-section, must be sheared in the outer row. The percentage strength of this single rivet is, from (104), $\%_1 \div n$. Hence:

$$\%_2 = 100 \cdot \frac{p-2d}{p} + \frac{\%_1}{n}. \tag{105}$$

The lowest of the values obtained from (103), (104), (105) is the percentage strength of the joint. An examination of these equations shows that, for double-strapped butt-joints, so long as d is not less than T, $\%_2$ is always greater than $\%$ or $\%_1$. This is also the case with lap-joints, so long as d is not less than

$$\frac{T}{\frac{23}{28} \times .7854} = \frac{T}{.64515}.$$

Since both of these conditions hold usually, the use of formula (105) will seldom be necessary.

3. DIAMETER AND PITCH. — The usual cases are:

(*a*) To find d so that $\% = \%_1$ (*i. e.*, equal tensile and shearing strengths of joint), when p, c, n, and T are given, equate (103) and (104):

$$100 \cdot \frac{p-d}{p} = 100 \cdot \frac{23A \cdot n \cdot c}{28p \cdot T}. \tag{106}$$

Substituting $A = \pi d^2/4$ and simplifying:

$$d^2 + \frac{1.55T}{n \cdot c} \cdot d = \frac{1.55p \cdot T}{n \cdot c};$$

$$d = \sqrt{\frac{.775T}{n \cdot c}\left(\frac{.775T}{n \cdot c} + 2p\right)} - \frac{.775T}{n \cdot c}. \tag{107}$$

(*b*) To find p, when d, c, n, and T are given, we have from (106):

$$p = \frac{23A \cdot n \cdot c}{28T} + d. \tag{108}$$

(*c*) To find d and p, when n, c, T, and $\% = \%_1$ are known, we have, from (103):

$$p = \frac{100}{100 - \%} \cdot d. \tag{109}$$

Substituting this value of p and $A = \dfrac{\pi d^2}{4} = \dfrac{22}{7} \cdot \dfrac{d^2}{4}$ in (104):

$$\frac{100 \times 23 \times 22 \times d^2 \cdot n \cdot c}{28 \times 7 \times 4 \times \dfrac{100d}{100 - \%} \cdot T} = \%_1 = \%;$$

$$d = \frac{1.55 \cdot \% \cdot T}{(100 - \%)n \cdot c}. \tag{110}$$

Substituting this value of d in (109):

$$p = \frac{155 \cdot \% \cdot T}{(100 - \%)^2 n \cdot c}. \tag{111}$$

As a rule, it will be found simpler to substitute the numerical value of d, as found from (110), in (109), thus obtaining a value for p directly, without using equation (111).

4. Diagonal Pitch and Width of Butt-Straps.—The resisting value of the net plate-section along the diagonal pitch, p_d, with regard to tensile stress normal to the joint, is usually—owing to the shearing component (§ 26) along the diagonal pitch—about $\frac{5}{6}$ of that of the same section, if located parallel to the joint, as in the pitch, p. Hence, the diagonal net section should be $\frac{1}{5}$ longer than

that part of the longitudinal section to which it should be equivalent in transverse tensile strength. In any event, the diagonal pitch should not be less than that found by the following formulæ:

(*a*) *Ordinary Zigzag Riveting and Chain Riveting with Alternate Rivets Omitted in the Outer Row.* Fig. 64 *c, d.* Reference to Fig. 64 *c*, shows that the same reasoning applies to both cases, since, in each, the net section of two diagonal pitches must be made equivalent in strength to the net section contained in the greatest longitudinal pitch, p. The liability of the plate to tear along A-B and B-D should be the same as along A-D or B-F. The net section along A-B-D is $2(p_d - d)T$ and that along A-D or B-F is $(p-d)T$. Hence:

$$2(p_d - d)T = \tfrac{6}{5}(p - d)T;$$

$$p_d = \frac{6p + 4d}{10}. \qquad (112)$$

From the right-angled triangle, A-B-C, we have, for the distance between the rows of rivets:

$$V = \sqrt{p_d^2 - \left(\frac{p}{2}\right)^2} = \sqrt{\left(\frac{6p + 4d}{10}\right)^2 - \frac{p^2}{4}};$$

$$V = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10}. \qquad (113)$$

The authors mentioned previously state that, for chain riveting the distance, V, should not be less than

$$\frac{4d + 1}{2}$$

and, as this result is greater than that obtained from (113), it is the one which, usually, is found tabulated.

The distance, E, from the centre of the nearest rivet-row, to the edge of plate, should not be less than $1.5\,d$. Hence, the minimum lap of plates in a lap joint or the half-width of butt-strap is:

$$2E + V. \qquad (114)$$

(*b*) *Zigzag Riveting with Alternate Rivets Omitted in Outer Row.* Fig. 73. The joint may fail by rupture of plate along A-B-C-D

or along A-D. For equality of strength, the resistances of the two net sections to tensile stress transverse to the seam, must be equal. The net section on B-C is $(p/2 - d)T$; that on A-D is $(p - d)T$; that on A-B or C-D is $(p_d - d)T$. The section, to which the metal left along A-B and C-D must be equivalent, is :

$$(p - d)T - \left(\frac{p}{2} - d\right) T = \frac{p}{2} \cdot T$$

$$\therefore 2(p_d - d)T = \tfrac{6}{5}\left(\frac{pT}{2}\right);$$

$$p_d = 0.3p + d. \qquad (115)$$

In the right-angled triangle, A-F-E, the base, F-$E = p/4$. For the distance, V_1, between the rows of rivets, we have :

$$V_1 = \sqrt{p_d^2 - \left(\frac{p}{4}\right)^2} = \sqrt{\left(\frac{3p + 10d}{10}\right)^2 - \left(\frac{p}{4}\right)^2}$$

$$V_1 = \frac{\sqrt{(11p + 20d)(p + 20d)}}{20}. \qquad (116)$$

As before, the lap or half-breadth of butt-strap is, with two rows of rivets :

$$2E + V_1. \qquad (117)$$

Since the riveting in all ordinary cases is of the form shown in Fig. 64, *c*, *d*, or Fig. 73, or a combination of those forms, formulæ (112), (113), (114), (115), (116) have a general application. For example, consider a double-strapped butt-joint, treble-riveted, zig-zag, with alternate rivets omitted in the outer row. The distance, V_1, between the outer and second row is obtained from (116); the distance, V, between the second and third rows is obtained from (113); and the half-breadth of butt-strap is :

$$2E + V_1 + V. \qquad (118)$$

5. Thickness of Butt-Straps. — To ensure strength and tightness, the aggregate thickness of the butt-straps should be more than that of the plate, since thin straps will bend and the joint will work and leak. For single butt-straps, $T_1 = \frac{9}{8}\ T$, and, for

double straps, $T_2 = \frac{5}{8} T$, are taken arbitrarily as the minimum values allowed, when no rivets are omitted.

When alternate rivets in the outer row are omitted, the empirical minimum thicknesses, as above, should be increased. Thus, with regard to joints shown in Figs. 64 *b*, *d*, let:

$T_2 =$ thickness of each butt-strap, no rivets omitted;
$T_2' =$ thickness of each butt-strap, alternate rivets omitted.

With no rivets omitted, the plate can tear from the strap or strap from plate in but one way, *i. e.*, along the net section of length, $p - d$. Hence, the ratio of strap-strength to plate-strength is:

$$\frac{2T_2(p-d)}{T(p-d)} = \frac{2T_2}{T}.$$

With alternate rivets omitted, the strap may tear from the plate along the inner and weaker half-pitch line but the plate cannot be ruptured along that line without also shearing one rivet in the outer row for each pitch-section. Hence, with the previous thickness, T_2, the strap would be weaker than the plate. The ratio of strap-strength to plate-strength, with thickness, T_2', is:

$$\frac{2T_2'(p-2d)}{T(p-d)}.$$

Equating the ratios and taking $T_2 = \frac{5}{8} T$:

$$\frac{2T_2}{T} = \frac{2T_2'}{T} \cdot \frac{p-2d}{p-d} \therefore T_2' = \frac{5}{8} T \cdot \frac{p-d}{p-2d}. \qquad (119)$$

Hence, when alternate rivets are omitted, the butt-straps must be thicker than when no rivets are omitted, in the ratio,

$$\frac{p-d}{p-2d}.$$

Under the same conditions, the thickness of a single butt-strap is

$$T_1' = \frac{9}{8} T \cdot \frac{p-d}{p-2d}. \qquad (120)$$

6. Joints Between Plates Unequal in Thickness. — In general, it is customary to proportion the joint as for two plates of

the smaller thickness. If the disparity be great, the proportions may be a mean proportional between those for the thick and those for the thin plate.

44. The Thickness of Shell Sheets.

The thickness of the shell plates of a cylindrical boiler depends upon the diameter, steam-pressure, tensile strength of plate, factor of safety, percentage strength of longitudinal joint, and allowance for corrosion. The shell is treated as a "thin cylinder" and its thickness is governed by the principles given in formulæ (1) to (4), § 1. Let:

$t =$ thickness of plate, ins.;
$r =$ internal radius of shell, ins.;
$P =$ steam pressure, gauge, lbs. per sq. in.;
$S_t =$ ultimate unit tensile strength of plate, lbs. per sq. in.;
$f =$ factor of safety;
$\% =$ percentage strength of longitudinal joint;
$\%/100 =$ strength of longitudinal joint as compared with solid plate.

If the shell were seamless and a factor of safety were not used, we would have by (4):

$$P \times r = t \times S_t \text{ and } t = \frac{P \cdot r}{S_t}.$$

The shell, however, is as strong only as its weakest part — the longitudinal joint. The strength of that joint is $\%/100$ times t. Hence, the thickness must be increased in inverse ratio over that required for a seamless shell and for t there must be substituted $t \times 100/\%$. Again, the working strength of the plate is equal to its ultimate strength, divided by the factor of safety. Therefore S_t must be replaced by $S_t \div f$. Making these substitutions:

$$\frac{t \times 100}{\%} = P \times r \times \frac{f}{S_t};$$

$$t = \frac{P \cdot r \cdot f}{\frac{\%}{100} \cdot S_t} \qquad (121)$$

If the joint be designed, as is usual, so that it will yield along the net plate-section of greatest pitch, we have, by (95), $\frac{\%}{100} = \frac{p-d}{p}$ and the equation becomes:

$$t = \frac{P \cdot r \cdot f}{\frac{p-d}{p} \cdot S_t} = \frac{P \cdot r \cdot f \cdot p}{(p-d)S_t}. \qquad (122)$$

For any given conditions, the value of % is known in close approximation or can be taken from Traill's tables. This value, substituted in (121), gives t at once.

The thickness may also be computed from (122) by substituting the value of p and d. Thus, assuming that the longitudinal seam is a butt-joint with double straps of equal width, we have from § 42:

$$p = \frac{S_c}{S_t} \cdot nd + d; \qquad (94)$$

$$p - d = \frac{nS_c}{S_t} \cdot d.$$

Substituting the values of p and $p - d$ in (122):

$$t = \frac{P \cdot r \cdot f}{n \cdot S_c}\left(\frac{n \cdot S_c}{S_t} + 1\right), \qquad (123)$$

in which n will be known from the type of joint and S_c and S_t by the tests of the plate. To provide for corrosion, $\frac{1}{16}$ inch is usually added to the value of t, as calculated, for the thick sheets of marine boilers.

45. The Stresses in Riveted Joints.

The riveted joint is not homogeneous and rigid, but is a built-up and, under stress, a comparatively yielding structure. The rivets, through which the load is transmitted, are not at first in contact with the walls of the rivet-holes, and the metal of both rivets and plates is not only elastic but will become, under excessive stress, plastic. "Lost motion," elasticity, and plasticity cause, with increasing pressure upon the joint, a continuous change in the load, form, and relative position of each element of the structure.

Thus, in the lap-joint, Fig. 77, when first riveted, the plates are pressed together by the axial contraction of the rivet in cooling, while the radial shrinkage of the latter leaves a slight annular space between its shank and the walls of the hole. The tension on the plates forms a couple whose arm is approximately t. When such

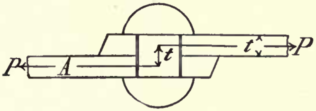


FIG. 77.

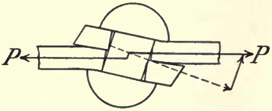


FIG. 78.

a joint is loaded, the couple tends to bring the plates into the same plane, thus bending the lap and inclining the rivet, as shown in Fig. 78. This action is opposed by the resistance of the plates to bending and by their friction at the contact-surfaces. When this friction is overcome, the plates slip on each other, the rivet bears at diagonal corners of the hole, and crushing pressure upon the walls of the latter, and shearing and bearing actions on the rivet are added to the tensile and bending stresses already existing in the plates and rivets. With an increasing load, these conditions grow in intensity but are modified in their local effect by the elasticity of the metal. Finally, in more or less of the elements of the joint, the plastic stage is reached and the rearrangement of stresses becomes more marked. At any time during these stages, failure may occur, if any element be strained beyond its ultimate strength.

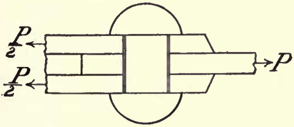


FIG. 79.

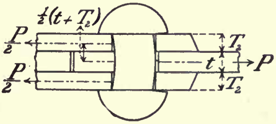


FIG. 80.

Similarly, in the double-strapped butt-joint, Fig. 79, there is at first no bearing of the rivet-shank upon plate or straps. The gradually applied load expends its force first in overcoming the frictional resistances between the plates and straps and between the latter and the rivet-heads. When slip at these surfaces occurs, the conditions are as in Fig. 80. The rivet is bent, it bears at one side on the plates and at the other upon the straps, and it is subjected to

double shear, while a bending moment, similar to that in a lap-joint, acts upon the straps. All stresses now prevail; and, with increasing load, the elasticity and ultimate plasticity of the metal will cause, until destruction, a continuous change in the stress upon any given element of the joint.

1. TENSILE STRESS IN RIVETS. — This stress is due to the contraction in cooling. Its magnitude — if the plates be tightly clamped during riveting — depends upon the coefficient of linear expansion, the modulus of elasticity, and the temperature at which riveting is completed. Let:

A = sectional area of rivet-shank;
l = length of rivet-shank;
E = modulus of elasticity;
τ = difference between temperature of cold rivet-blank and that at which riveting is completed;
λ = increase, total, in length for change of τ°;
s = increase, per unit of length, for change of τ°;
S_t = unit tensile stress produced by contraction;
a = coefficient of linear expansion for a change of 1° F.

Then:

$$\lambda = a \cdot \tau \cdot l ; s = \frac{\lambda}{l} = a \cdot \tau ; E = \frac{S_t}{s} ;$$

$$S_t = E \cdot s = a \cdot \tau \cdot E.^* \qquad (124)$$

The total tensile load on the shank $= A \times S_t$. Equation (124) shows that this load is independent of the length of the shank. The deduction holds only while S_t lies within the elastic limit of the metal.

For steel in general, $a = .0000065$ and $E = 30{,}000{,}000$; and, for soft rivet steel, S_t at the elastic limit = 30,000 lbs. per sq. in. These values, substituted in (124), give $\tau = 154°$. In practice, however, the range of temperature greatly exceeds this. Hence, it follows that, with good workmanship, the cooling and attempted contraction strain the shank considerably beyond the elastic limit.

The actual tension in the shank is uncertain, since permanent set is produced and the elasticity of the metal is impaired. Mr. Stoney, in experiments quoted previously (§36), gives the contractile strength of iron rivets with hand-made points as 12.32 tons per sq. in. of rivet-section, at which stress the points or heads

*Merriman: "Mechanics of Materials," 1899, p. 145.

flew off. He gives also, from his experiments, 0.6 as the coefficient of friction of ordinary steel plates. For the latter, with rivets as above, the frictional resistance to slip would then be $12.32 \times 0.6 = 7.39$ tons $= 16,553$ lbs. per sq. in. of rivet-section. Professor Bach's experiments * show, for good single lap-riveting, similar values ranging from 14,000 to 25,000 lbs. per sq. in.

2. BENDING STRESS ON RIVETS, Figs. 77 to 80, inclusive. In lap-riveting, this stress is a maximum as soon as the plates engage the rivet, which position is approximately that shown in Fig. 77. The rivet, in the lap-joint, acts as a cantilever, the distance between load and reaction being t; in the double-strapped butt-joint this distance is $(t + T_2)/2$ and the rivet is a simple beam, loaded in the middle. The classification and distances are approximate and must be regarded as simply an indication of the principle involved. Let:

$P =$ total load on rivet;

$\frac{I}{c} =$ section-modulus of rivet $= \frac{\pi d^3}{32}$;

$S =$ maximum stress, tensile or compressive, due to bending;

$M =$ maximum bending moment;

$= P \cdot \frac{t + T_2}{4}$ in butt-joints, double-strapped;

$= P \cdot t$ in lap-joints;

$S \cdot \frac{I}{c} =$ resisting moment.

Equating the bending and resisting moments, we have for:

$$\textit{Lap-joints:} \quad S = \frac{32 P \cdot t}{\pi d^3};$$

$$\textit{Butt-joints:} \quad S = 8P \cdot \frac{t + T_2}{\pi d^3}.$$

In the lap-joint, as the plates bend, the bending moment decreases.

3. SHEARING STRESS ON RIVETS. — When failure by shearing occurs in lap and single-strapped butt-joints, but one cross-section of the rivet-shank is sheared, while, in double-strapped butt-

* "Die Maschinen-Elemente," 1901, p. 170.

joints, the shank must be sheared in two places. In the former case, the rivet is said to be in single, in the latter in double, shear. The resistance to shearing is the product of the area sheared by the *mean* ultimate shearing stress; or, in the case of a rivet, $\pi d^2/4 \times S_s$, for each rivet-section.

In double-strapped joints, as shown in Fig. 80, the shearing force, P, is always approximately normal to the rivet. Hence, the shearing resistance of the latter is $2(\pi d^2/4 \cdot S_s)$. In the lap-joint, Fig. 78, as the laps bend, the force, P, does not remain normal and, hence, has both shearing and tensile components with regard to the axis of the rivet. The mean ultimate shearing strength of steel is taken as 0.8 of its mean ultimate tensile strength. Therefore, the division of the force, P, in single shear, into tensile and shearing components diverts a portion of it to the stress against which the rivet's resistance is greater. As a consequence, the ultimate strength, in joints, of a rivet in single shear is to that of one in double shear, not as 1 : 2, but as 4 : 7 or 1 : 1.75, approximately. For each rivet in a joint, then:

$$\textit{Ultimate strength, single shear} = \frac{\pi d^2}{4} \cdot S_s;$$

$$\textit{Ultimate strength, double shear} = \frac{\pi d^2}{4} \cdot S_s \times 1.75,$$

in which d is the diameter of the shank and S_s is the *mean* ultimate unit shearing stress.

The shearing stress varies in intensity throughout the cross-section. In a solid of circular section, as a rivet-shank, the maximum is $\frac{4}{3}$ the mean shearing stress.* Taking the *maximum* shearing unit stress, S_s(max.), as $\frac{8}{10}$ of the ultimate unit tensile stress, S_t, the *mean* ultimate shearing stress,

$$S_s = \tfrac{3}{4} S_s(\text{max.}) = \tfrac{3}{4} \times \tfrac{8}{10} S_t = \tfrac{3}{5} S_t.$$

While this is the theoretical ratio between the maximum and mean shearing stresses of a solid circular section, the common practice is to take $S_s = 0.8 S_t$ for steel rivets, the discrepancy, if any, being covered by the factor of safety.

4. Bearing Stress on Rivets and on the walls of rivet-holes.

* Rankine: "Applied Mechanics," 1869, p. 340.

In a new joint, the rivet-shank, owing to contraction in cooling, is not in contact with the plate, and, further, its initial tension produces frictional resistance to movement between the plates, straps, and rivet-heads. This friction, when the joint is first loaded, reduces the pressure upon the rivet and it must be overcome wholly before the full bearing stress can exist. In fact, if it be assumed that each rivet carries the same load, there would be, in service and with the customary factor of safety, no bearing pressure whatever upon the rivets of a new joint. Thus, taking the mean ultimate shearing stress of rivet-metal as 44,000 lbs. per sq. in., the factor of safety as 4.5, the elastic limit as 30,000 lbs. per sq. in., and the coefficient of friction of steel plates as 0.5, we have, per sq. in. of rivet-section:

Allowable shearing load $= 44{,}000 \div 4.5 = 9{,}778$ lbs.;

Frictional resistance at elastic limit $= 30{,}000 \times 0.5 = 15{,}000$ lbs.,

i. e., even if the elastic limit were not exceeded, the resistance of the plates to slip would be 50 per cent. greater than the permissible shearing load. In view of these considerations, high authorities in France and Germany (§46) contend that riveted joints should be designed with regard to their frictional resistance and not with reference to the ultimate strength of their elements.

On the other hand, owing to the elasticity of the plate, the irregularity of workmanship, the bending of plates or straps, and their actual and relative movement, especially in pressure-joints, during expansion and contraction, it seems probable that frictional resistance is modified greatly in service. Furthermore, experiment shows that, in multiple riveted joints, the outer lines of the rivets in test-specimens bear a wholly disproportionate share of the load, owing doubtless to the elasticity of the plate between them and the inner rows. Hence, their load probably exceeds the frictional resistance they produce and the latter may be overcome in detail throughout the joint.

In considering bearing pressure, assume for simplicity, as in Fig. 81, that the rivet is incompressible, that the elastic limit of the metal is not exceeded, and that the rivet axis remains parallel to the walls of the hole. The total load, P, upon the rivet will force the latter into the plate the distance, $O\text{-}O' = A\text{-}A'$, since the

displacement, parallel to the line of action of P, will be the same for all compressed parts of the plate in front of the rivet. The elasticity and reaction of the plate produce on any element, B, of the circumference a bearing pressure, b, which is a maximum, b', at C and varies as the cos θ throughout the quadrant, being zero at D. The summation of the vertical components of b equals P.

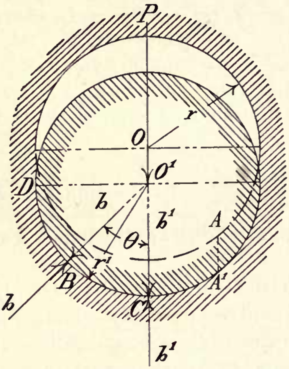


FIG. 81.

The rivet, however, is compressible and its section under pressure is no longer circular. Again, Figs. 78 and 80 show that the pressure on the axial plane is not uniform throughout. Finally, under excessive stress, the plastic stage is reached and the intensity and distribution of the pressure depend upon the freedom of flow of the metal. These unknown elements make an analysis of the problem impossible without assumptions so broad as to render the results valueless. With regard to an empirical formula, it may be noted that, in summation by the calculus of the vertical components of b, the normal pressure upon the elementary arc, $ds = r'd\theta$, must be considered and that the radius, $r' = r = d/2$, approximately. Again, with a given load, P, the resulting unit bearing pressure depends, in some degree, upon the thickness, t. These conditions warrant the introduction of d and t in such a formula, the latter becoming:

$$P = S_c \times d \times t, \tag{125}$$

in which $P=$ safe load on one rivet and $S_c=$ a mean working bearing stress determined by experiment. Professor Unwin* says:

From experiments on indentation, it is known that the resistance to indentation of a plastic material does not much depend on the form of the indenting body, but only on the projected area normal to the direction of indentation. Hence, it is not an arbitrary rule, but one based on experiment, to take the resistance to indentation of a plate of thickness, t, by a rivet of diameter, d, to be proportional to the projected area, $d \times t$.

5. BENDING STRESS IN PLATES.—In a lap-joint, the stress will be a maximum when the parts are in the position shown in Fig. 77. For example, in plate, A, there acts at the left a force, P, which produces a direct unit tensile stress, S_t, and, at the right, an opposing force, P, with leverage, t, which tends to bend the plate and to produce a further tensile stress, S_b, in its upper fibres. The breadth of the section thus bent is $p-d$ and its depth is t. The stresses are:

Lap-Joint (Fig. 77):

Tensile load on section $= P$;

Resistance of section $= S_t(p-d)t$;

Equating load and resistance: $S_t = \dfrac{P}{(p-d)t}$.

Bending moment of load $= P \times t$;

Modulus of section $= \dfrac{(p-d)t^2}{6}$;

Resisting moment of section $= S_b \cdot \dfrac{(p-d)t^2}{6}$;

Equating the moments: $S_b = \dfrac{6P}{(p-d)t}$.

Total maximum unit tensile stress $= S_t + S_b$;

$$= \frac{7P}{(p-d)t}. \qquad (126)$$

Double-strapped Butt-joint (Fig. 80):

Tensile load on strap $= \dfrac{P}{2}$;

* "Elements of Machine Design," Part I., p. 131, 1901.

Resistance of strap $= S_t(p-d)T_2$;

Equating load and resistance: $S_t = \dfrac{P}{2(p-d)T_2}$.

Bending moment of load $= \dfrac{P}{2}\cdot\dfrac{t+T_2}{2}$;

Modulus of section $= \dfrac{(p-d)T_2^2}{6}$;

Resisting moment $= S_b\cdot\dfrac{(p-d)T_2^2}{6}$;

Equating the moments: $S_b = \frac{3}{2}P\cdot\dfrac{t+T_2}{(p-d)T_2^2}$.

Total maximum unit tensile stress

$$= S_t + S_b = \frac{P}{2(p-d)T_2}\left(1 + \frac{3(t+T_2)}{T_2}\right). \quad (127)$$

These calculations will be regarded as general, giving maximum results. In lap-joints especially, the moment of the load is reduced rapidly by the bending of the plates.

6. Bearing, Shearing, and Tensile Stresses in Plates.— The bearing pressure between the rivet and the walls of the rivet-hole, is, of course, mutual. The metal in front of the rivet is, as indicated in § 41, in the condition, approximately, of a beam fixed at the ends, with, in consequence, a shearing stress at the latter since, at those points, the stress due to the resistance of the rivet is communicated to the net section of plate along the pitch-line. The distribution of the tensile stress in this net section is affected by several conditions and presents a complex problem which Mr. C. E. Stromeyer * contends is treated most adequately by regarding the metal surrounding the rivet as part of a section of a thick-walled cylinder.

It is possible to gain from experiments some knowledge of the conditions which prevail. Thus, when a rectangular specimen of *unperforated* plate is tested to rupture, the fracture is of crescent form and, if the separated parts be brought together, they will touch at the sides, leaving a gap in the middle. Evidently, then, the stress is a maximum in the centre of the specimen. Again,

* "Marine Boiler Management and Construction," 1893, p. 166.

as shown in §38, the perforation of a plate, as for rivets, causes a restriction of the flow of metal, a change in stress-distribution, and an increase in ultimate tenacity, owing apparently to the partial removal of stress from the centre of the net plate-section to the portions adjoining the holes. Furthermore, when such a plate forms part of a loaded joint, it is the rivets which produce stress in it. If the plate be considered, very generally, as made up of simple beams, each loaded in the centre by one rivet, the stress would be a maximum at the walls of the hole and reach its minimum at the centre of the net plate-section.

46. The Friction of Riveted Joints.

Many tests to determine the resistance to slip of riveted joints of various types, have been made at the U. S. Arsenal, Watertown, Mass., the results of which will be found in the various annual reports to the Secretary of War. M. Dupuy,* also, has conducted extensive experiments with regard to the magnitude and effect of the friction of the joint. The researches of Professor C. Bach, of Stuttgart, have been exhaustive and he has presented strong argument in favor of the theory which bases the design of riveted joints upon their frictional resistance alone. The review of his conclusions which follows, has been summarized from the matter, as set forth in his work on *Machine Design.*†

The effect, upon the frictional resistance, of the temperature, the length of the shank, the number of rivet-rows, etc., is considered separately. Unless otherwise stated, the joint was not calked. Consider:

1. THE TEMPERATURE AT RIVETING, either cherry-red or rose-red. Notation: $t =$ thickness of plate, $d =$ diameter of rivet, $l =$ length of shank. One kilogramme (*kg.*) = 2.20462 lbs., avoirdupois; one millimetre (*mm.*) = 0.03937 in.; one square centimetre (*qcm.*) = 0.155 sq. in.

(*a*) Lap-joint; $t = 13$ *mm.*, $d = 19$ *mm.*, $l = 26$ *mm.* The lower (cherry-red) temperature gave sometimes a greater friction than the higher, averaging 1,199 to 1,115 *kg.* per *qcm.* of rivet cross-section.

* An. d. Ponts et Chaussées.

† "Die Maschinen-Elemente," 1901, pp. 165–170.

(*b*) Lap-joint with inside and outside welt-strips; $t = 13$ *mm.*, $d = 19$ *mm.*, $l = 52$ *mm.*, t (each welt) $= 13$ *mm.* Owing to the doubled length of shank, there was given, at the higher temperature, a greater friction, averaging 1,769 to 1,305 *kg.* per *qcm.* of rivet cross-section.

Experiments by Considère confirm (*a*) and apparently contradict (*b*). He found, that, at a riveting temperature of 600° to 700° C., the friction reached a maximum, being then greater than when the rivet was at a rose-red heat, say 1000° C. Prof. Bach concludes that it is not the temperature of the rivet at insertion which is important but that at the moment of finishing the point. Also, in machine-riveting, his experiments showed, that, within limits, the friction increased with the duration of the pressure upon the rivet. The resistance was affected further by the temperature, at finishing, of the portions of plate adjacent to the rivet.

2. LENGTH OF RIVET-SHANK. — The shank of greater length produced the higher resistance. Thus:

(*a*) Lap joint; $t = 7.5$ *mm.*, $d = 16$ *mm.*, $l = 15$ *mm.* Resistance, 846 *kg.* per *qcm.* of rivet cross-section.

Lap joint; $t = 7.5$ *mm.*, $d = 16$ *mm.*, $l = 31$ *mm.* Resistance, 1,037 to 1,111 *kg.* per *qcm.* of rivet cross-section.

(*b*) Lap joint; $t = 13$ *mm.*, $d = 19$ *mm.*, $l = 26$ *mm.* Resistance, 1,115 *kg.* per *qcm.* of rivet cross-section.

Lap joint; $t = 13$ *mm.*, $d = 19$ *mm.*, $l = 52$ *mm.* Resistance, 1,769 *kg.* per *qcm.* of rivet cross-section.

The slip with varying loads is shown by the following experiments:

(*a*) Single-riveted lap-joint; $t = 13$ *mm.*, $d = 19$ *mm.*, $l = 26$ *mm.*, pitch $= p = 48$ *mm.*, number of rivets $= n = 3$, diameter of rivet-hole $= 20$ *mm.*

Load.	Load on Rivet Area.	Slip.
10,000 *kg.*	1,174 *kg.* per *qcm.*	0
11,000 "	1,291 "	0.0125 *mm.*
15,000 "	1,761 "	0.1 "
20,000 "	2,348 "	1.175 "

(*b*) Same joint as (*a*), excepting that length of shank $= 52$ *mm.*, a plate, 13 *mm.* thick, having been laid on each side of the seam.

It will be seen that the resistance of (*b*) was 15,000 *kg.*, while that of (*a*) was 10,000; but that, with (*b*), the slip increased far more rapidly, and, at 20,000 *kg.* load, was much greater.

Load.	Load on Rivet Area.	Slip.
15,000 *kg.*	1,761 *kg.* per *qcm.*	0.
16,000 "	1,878 "	0.004 *mm.*
17,000 "	1,995 "	0.009 "
18,000 "	2,113 "	0.202 "
19,000 "	2,230 "	1.255 "
20,000 "	2,348 "	1.405 "

The breaking strengths of these joints were:

	Kg. per *qcm.*, Rivet-section.
(*a*)	3,522
(*b*)	3,404

Prof. Bach's conclusions as to the greater length of rivet-shank giving the greater resistance are, apparently, at variance with the theory of contractile stresses, as given in § 45. Assuming absolutely the same conditions throughout, excepting dissimilar aggregate plate-thicknesses and shank-lengths, the contractile-stress, pressure, and frictional resistance per sq. in. of rivet-section should be the same in all cases, since $S_t = \alpha \cdot \tau \cdot E$ in (124). The explanation of this seeming discrepancy lies probably in the fact that, with the shorter shank, the expansion, while the same percentage of the length, is a less amount actually; and, therefore, in riveting, will be more affected by the same looseness of plates or other defect, than the expanded length of the longer shank.

3. Number of Rows of Rivets. — The frictional resistance to slip does not increase proportionately in passing from single to multiple riveting, owing to the fact that the elasticity of the plate prevents the regular and proportionate distribution of the load upon the joint. Thus, in a lap-joint, chain-riveting, 6 rows, $t = 12$ *mm.*, $d = 19$ *mm.*, diameter of rivet-hole = 20 *mm.*, width of plate = 150 *mm.*, in the plane of the cross-section of the rivets slip, was observed at

6,000 *kg.*,	load in	1st	and	6th	rows.
8,000 "	"	2d	"	5th	"
11,000 "	"	3d	"	4th	"

The slip, therefore, was greater in the outside rows, owing to the unequal distribution of the load.

4. Double-Strapped Butt-Joints. — The single-riveted butt, as compared with the single-riveted lap-joint, gives a somewhat less resistance. Thus, t(plate) = 13 to 14 *mm.*, t(strap) = 9 *mm.*,

$d = 19$ *mm.*, resistance $= 906$ *kg.* per *qcm.*, while, in the single-riveted lap-joint, with $t = 12.5$ *mm.* and $d = 19$ *mm.*, the resistance $= 1,186$ *kg.* per *qcm.* This difference arises from the absence of plate-bending in the butt-joint, the plates and load being in the same plane, while, in the lap-seam, the eccentricity of the load, in bending the plates, clamps them more closely and gives greater friction. Multiple riveting, in butt-joints, is affected by the elasticity of the plate in a manner similar to that which has been described for multiple lap-riveting.

5. Machine Riveting. — In machine-riveting, the magnitude of slip-resistance depends greatly upon the duration of pressure upon the rivet. With rapid work, the friction may be less than in hand-riveting. With sufficient pressure and duration the reverse is true, especially when thick plates and, consequently, large rivets are employed.

6. Influence of Calking. — The effect of calking is shown, in detail, by the results, as follows, of experiments upon 25 lap-joints, in each of which $t = 12$ *mm.*, $d = 19.5$ *mm.*, diameter of holes $= 20.5$ *mm.* The holes were drilled and the joints hand-riveted.

		Plates.	Rivet-Heads.
(*a*)	5 Joints.	Not Caulked.	Not Calked.
(*b*)	5 "	Caulked both sides.	" "
(*c*)	5 "	" one side.	Calked one side.
(*d*)	5 "	" both sides.	" " "
(*e*)	5 "	" " "	" both sides.

The results, in *kg.* per *qcm.* of rivet cross-section, were:

	Resistance to Slip.	Breaking Load.
(*a*)	881	3,397
(*b*)	1,238	3,413
(*c*)	1,327	3,311
(*d*)	1,572	3,178
(*e*)	1,617	3,258

Other things being equal, a greater proportional advance in frictional resistance will be made by calking the heads of a short, then a long, rivet, since that resistance depends also upon the length of the rivet-shank.

7. Résumé.— Prof. Bach's experiments show that, (*a*) in good single lap-riveting, there will be a frictional resistance ranging from 1,000 to 1,800 *kg.* per *qcm.* of rivet cross-section, or even

more, according to length of shank and width of specimen tested; (*b*) the age of the joint has an appreciable effect upon the amount of resistance; (*c*) the magnitude of the resistance is fully adequate to transmit the load generally placed upon a riveted joint; (*d*) calking increases considerably the resistance, a fact which is of importance, not only in pressure-joints but also in structural work in cases where inaccessibility makes good riveting difficult.

CHAPTER IV.*

RIVETED JOINTS: TESTS AND DATA FROM PRACTICE.

47. Tests of Multiple-Riveted, Double-Strapped Butt-Joints.

The tests whose records follow were conducted under the supervision of the Ordnance Department, U. S. Army, at the Watertown Arsenal in 1887, for the Bureau of Steam Engineering, U. S. Navy. They are of especial value, since the specimens were unusually wide, the plates thick, and multiple riveting was used, the conditions thus corresponding with those of boiler-joints for moderate pressures. The plates were of open-hearth steel with drilled holes and sheared edges and the joints were riveted by steam. The strips tested to show the quality of the metal were of the same grade, although not from the same sheets, as the joints. The mean tensile strength of three such specimens for each thickness was used in computing the efficiencies of the joints. The plate-thickness varied somewhat at different edges. After testing, the rivet heads were planed from a number of the joints, the rivets driven out and butt-straps removed, and the elongation of the rivet-holes measured. Owing to the absence of tensile tests of the unperforated plate, the efficiencies of joints I_1, I_2, I_3 were not computed.

The unequal distribution of the load among the various rows of rivets is shown clearly by the elongations of the rivet-holes. Thus, in the joint, B_2, Fig. 82 ($\frac{5}{8}$-in. plate, $\frac{3}{4}$-in. steel rivets) which failed by rupture at 50,200 lbs. apparent tension on net plate section, the average elongations on the right of the seam were, in the outer row, 0.284 in.; in the central row, 0.173 in.; and, in the inner row, 0.054 in. Again, the average elongation of the two

* For the data from practice given in this chapter, the author is indebted to the Bureau of Steam Engineering, U. S. Navy; the Baldwin Locomotive Works; Messrs. R. D. Wood and Company; J. M. Allen, Esq., President, The Hartford Steam Boiler Inspection and Insurance Company; E. D. Meier, Esq., of the American Boiler Manufacturers' Association; the Editor of the *American Machinist;* C. C. Schneider, Esq., Vice-President, American Bridge Company; the Bureau of Construction and Repair, U. S. Navy; Edwin S. Cramp, Esq., Vice-President, the William Cramp Ship and Engine Building Company; and W. Irving Comes, Esq., Secretary, the American Bureau of Shipping.

end holes in the outer row was 0.31 in.; that of the middle hole, same row, was 0.26 in. Similar values for the central row were 0.22 in, and 0.145 in.; and, for the inner row, 0.065 in. and 0.05 in. The stress in the joint section was, therefore, greatest at the edges.

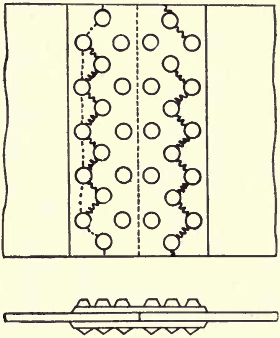


FIG. 82.

As shown in Fig. 82, the metal drew down in thickness, in diagonal and zigzag lines between the rivet-holes of adjacent rows, although the space thus traversed was greater than that along the pitch-line. All riveting, through two butt-straps, was zigzag with no rivets omitted.

In the tables which follow, tests No. 905 and 909 are sample records of the strips tested to show the quality of the metal. The succeeding tables give tension tests of metal from the fractured ends of the joints, the tests of the joints, and data as to the mode and appearance of fracture. The widths of the various classes of joint-specimens tested, were:

A, *B*, *K*, 20 ins.; *D*, 17 ins.; *E*, 16.5 ins.; *G*, 15.75 ins.; *H*, 20.12 ins.; *I*, 14.39 ins.

TEST NO. 905.—SPECIMEN C_2.—THICKNESS, FIVE EIGHTHS INCH.

Gauged length, 15 inches; cross section, 12″ × .639″; area, 7.668 square inches.

Applied Loads.		In Gauged Length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,668	1,000	0.0000	0.0000	Initial load.
38,340	5,000	.0017	0.0000	
76,680	10,000	.0042	0.0000	
115,020	15,000	.0065	0.0000	
153,360	20,000	.0089	0.0000	
161,028	21,000	.0094		
168,696	22,000	.0098		
176,364	23,000	.0103		
184,032	24,000	.0108		
191,700	25,000	.0112	—.0001	
194,000		.0113		
196,000		.0115		
198,000		.0116		
200,000		.0118		
202,000		.0120		
204,000		.0123		
206,000		.0125		
208,000		.0127		
210,000		.0130		
212,000		.0133		
214,000		.0136		
216,000		.0140		
218,000		.0144		
220,000		.0150		
222,000	28,950	.0154		Elastic limit.
224,000		.0162		
226,000		.0170		
228,000		.0183		
230,000		.0213		
232,000		.0277		
234,000		.0480		
236,000		.0995		
237,708	31,000	.2175		
245,376	32,000	.23		
253,044	33,000	.25		
260,712	34,000	.28		
268,380	35,000	.31		
276,048	36,000	.33		
283,716	37,000	.37		
291,384	38,000	.41		
299,052	39,000	.45		
306,720	40,000	.50		
314,388	41,000	.54		
322,056	42,000	.59		
329,724	43,000	.66		
337,392	44,000	.71		
345,060	45,000	.78		
352,728	46,000	.86		
360,396	47,000	.95		
368,064	48,000	1.07		
375,732	49,000	1.21		
383,400	50,000	1.47		
391,068	51,000	1.58		
398,736	52,000	1.80		
406,404	53,000	2.08		
414,072	54,000	2.98		
414,800	54,100	3.42		Tensile strength.
0	0	0	5.52	= 36.8 per cent.

Elongation of inch sections: .23″, .20″, .24″, .26″, .34″, .35″, .40″, *1.03″, .69″, .48″, .35″, .31″. 26″, .21″, .17″.

Area at fracture, 9.62″ × .48″ = 4.618 square inches. Contraction, 39.8 per cent.

Appearance of fracture silky, slightly lamellar. Fracture open at middle, .38″, at edges, .10″ and 0.

TEST NO. 909.—SPECIMEN F_3.—THICKNESS, SEVEN EIGHTHS INCH.

Gauged length, 15 inches; cross section, 8.510″ × .867″; area, 7.378 square inches.

Applied loads.		In gauged length.		Remarks.
Total.	Per square inch.	Elongation.	Set.	
Pounds.	*Pounds.*	*Inches.*	*Inches.*	
7,378	1,000	0.0000	0.000	Initial load.
36,890	5,000	.0020	0.000	
73,780	10,000	.0045	0.000	
110,670	15,000	.0070	0.000	
147,560	20,000	.0097	0.000	
154,930	21,000	.0103		
162,316	22,000	.0108		
169,694	23,000	.0115		
172,000		.0120		
174,000		.0122		
176,000		.0124		
178.000		.0127		
180,000		.0130		
182,000		.0134		
184,000		.0137		
186,000		.0142		
188,000		.0147		
190,000	25,750	.0153		Elastic limit.
192,000		.0165		
194,000		.0182		
196,000		.1775		
206,584	28,000	.23		
213,962	29,000	.26		
221,340	30,000	.28		
228,718	31,000	.32		
236,096	32,000	.35		
243,474	33,000	.38		
250,852	34,000	.43		
258,230	35,000	.47		
265,608	36,000	.52		
272,986	37,000	.57		
280,364	38,000	.63		
287,742	39,000	.69		
295,120	40,000	.74		
302,498	41,000	.84		
309,876	42,000	.92		
317,254	43,000	1.01		
324,632	44,000	1.16		
332,010	45,000	1.28		
339,388	46,000	1.47		
346,766	47,000	1.67		
354,146	48,000	1.98		
361,524	49,000	2.48		
365,700	49,570	3.30		Tensile strength.
0	0		5.31	= 35.4 per cent.

Elongation of inch sections: .17″, .20″, 22″, .25″, .29″, .36″, .49″, *1.41″, .51″, .37″, .29″, .25″, .21″, .17″, .12″.

Area at fracture, 6.43″ × .58″ = 3.73 square inches. Contraction, 49.4 per cent.

Appearance of fracture, silky, lamellar. Fracture open at the middle, .40″; edges closed.

Table XLI.

Tests of Riveted Joints for Bureau of Steam Engineering,

No. of Test.	Mark on Piece.	Style of Joint.	Nominal Thickness of Plate.	Size and Kind of Rivets and Holes.	Sectional Area of Plate. Gross.	Sectional Area of Plate. Net.
			Inch.		Sq inches.	Sq inches.
910	A_1	Double riveted; double butt straps $\frac{1}{2}$ in. thick; 3 in. pitch.	$\frac{5}{8}$	$\frac{3}{4}$ inch steel rivets; $\frac{25}{32}$ inch drilled holes.	13.22	10.12
911	A_2		$\frac{5}{8}$		13.08	10.01
912	A_3		$\frac{5}{8}$		12.41	9.50
913	B_1	Treble riveted; double butt straps $\frac{1}{2}$ in. thick; $3\frac{9}{16}$ in. pitch.	$\frac{5}{8}$	$\frac{3}{4}$ inch steel rivets; $\frac{25}{32}$ inch drilled holes.	12.81	10.31
914	B_2		$\frac{5}{8}$		12.69	10.21
915	B_3		$\frac{5}{8}$		12.691	10.21
916	D_1	Double riveted; double butt straps $\frac{3}{4}$ in. thick; $3\frac{3}{4}$ in. pitch.	$\frac{7}{8}$	1 inch steel rivets; $1\frac{1}{16}$ inch drilled holes.	14.671	10.09
917	D_2		$\frac{7}{8}$		14.646	10.07
918	D_3		$\frac{7}{8}$		14.705	10.11
919	E_1	Treble riveted; double butt straps $\frac{3}{4}$ in. thick; $4\frac{9}{16}$ in. pitch.	$\frac{7}{8}$	1 inch steel rivets; $1\frac{1}{16}$ inch drilled holes.	14.322	10.63
920	E_2		$\frac{7}{8}$		14.438	10.72
921	E_3		$\frac{7}{8}$		14.256	10.58
922	G_1	Leavitt joint; double butt straps; one of usual width for double riveting and $\frac{7}{8}$ inch thick; other, $\frac{5}{8}$ inch thick and extended far enough on each side to receive five additional rivets in two rows. Pitch of double riveting on inner rows, $2\frac{5}{8}$ inch.	$\frac{7}{8}$	3 plates; $1\frac{1}{16}$ inch iron rivets; $1\frac{1}{8}$ in. drilled holes. 2 plates; $1\frac{1}{8}$ in. iron rivets; $1\frac{1}{4}$ inch drilled holes.	13.781	10.93
923	G_2		$\frac{7}{8}$		13.852	10.985
924	G_3		$\frac{7}{8}$		13.741	10.90
925	H_1	Leavitt joint; same arrangement as in G series except that wide butt strap has six rivets on each side beyond narrow strap. Pitch of double riveting, $2\frac{7}{8}$ inch.	$\frac{5}{8}$	3 plates; 1 inch iron rivets; $1\frac{1}{16}$ in. drilled holes. 2 plates; $1\frac{1}{8}$ inch iron rivets; $1\frac{1}{4}$ inch drilled holes.	13.252	10.537
926	H_2		$\frac{5}{8}$		12.776	10.154
927	H_3		$\frac{5}{8}$		12.81	10.186
928	I_1	Leavitt joint; same arrangement as in G series except that the five rivets in ends of wide butt strap are differently spaced. Pitch of double riveting on inner rows, $2\frac{7}{8}$ inch.	$\frac{9}{16}$	$\frac{15}{16}$ inch iron rivets; 1 inch drilled holes.	8.287	6.820
929	I_2		$\frac{9}{16}$		8.211	6.755
930	I_3		$\frac{9}{16}$		8.28	6.81
931	K_1	Treble riveted; double butt straps $\frac{1}{2}$ in. thick; $3\frac{9}{16}$ in. pitch.	$\frac{5}{8}$	$\frac{3}{4}$ inch iron rivets; $\frac{25}{32}$ inch drilled holes.	12.36	9.94
932	K_2		$\frac{5}{8}$		12.93	10.40
933	K_3		$\frac{5}{8}$		12.98	10.44

* No figures given because no tests were made of this thickness of metal for tensile strength.

TABLE XLI.—*Continued.*

UNITED STATES NAVY DEPARTMENT.

Bearing Surface of Rivets.	Shearing Area of Rivets.	Tensile Strength of Plate per sq. Inch.	Maximum Stress on Joint per Square Inch.					
			Tension on Gross Section of Plate.	Tension on Net Section of Plate.	Compression on Bearing Surface of Rivets.	Shearing on Rivets.	Efficiency of Joint.	
Sq. inches.	*Sq. inches.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Per cent.*	
6.72	12.46	53,710	42,860	55,990	84,320	**45,470**	79.8	a
6.64	12.46	53,710	40,960	**53,530**	80,690	43,000	76.2	b
6.30	12.46	53,710	42,720	**55,800**	84,140	**42,540**	79.5	c
8.01	15.34	53,710	43,460	**54,040**	69,560	36,320	80.9	d
7.93	15.34	53,710	40,390	**50,200**	64,630	33,410	75.2	e
7.94	15.34	53,710	44,290	**55,050**	70,790	36,640	82.5	f
8.25	15.96	51,190	35,180	**51,150**	62,560	32,340	68.7	g
8.23	15.96	51,190	36,190	**52,640**	64,410	33,210	70.7	h
8.27	15.96	51,190	35,780	**52,050**	63,630	32,970	69.9	i
10.15	19.51	51,190	37,910	**51,080**	53,500	27,830	74.1	j
10.23	19.51	51,190	38,400	**51,720**	54,190	28,420	75.0	k
10.10	19.51	51,190	37,950	**51,130**	53,560	27,730	74.1	l
16.30	28.00	51,190	40,810	**51,460**	34,500	20,090	79.7	m
16.38	28.00	51,190	41,740	**52,640**	35,300	20,650	81.5	n
16.26	28.00	51,190	40,120	**50,580**	33,910	19,690	78.4	o
14.045	30.41	53,710	42,820	**53,860**	40,410	18,660	79.7	p
13.548	30.41	53,710	45,300	**57,000**	42,720	19,030	84.3	q
13.576	30.41	53,710	46,070	**57,940**	43,470	19,410	85.8	r
8.057	18.06	*	42,250	**51,330**	43,450	19,380	*	s
7.994	18.06	*	40,800	**49,590**	41,910	18,550	*	t
8.05	18.06	*	40,720	**49,520**	41,890	18,670	*	u
7.773	15.34	53,710	43,600	**54,220**	69,720	**35,130**	81.2	v
8.08	15.34	53,710	43,260	**53,780**	69,220	36,460	80.5	w
8.11	15.34	53,710	43,000	53,460	68,820	**36,390**	80.1	x

Figures in heavy-faced type indicate manner of failure.

For explanation of reference letters in last column, see next page.

MODE OF FRACTURE AND APPEARANCE OF FRACTURED SURFACES.

a. Sheared the rivets in one plane in Plate A; started a fracture at side of one rivet hole in outside row of riveting.

b. Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, lamellar.

c. Fractured Plate A along outside row of rivet holes; sheared (double shear) six rivets in Plate B. Appearance of fractures, silky, slight lamination.

d. Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky.

e. Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, slightly lamellar.

f. Fractured Plate A along outside row of rivet holes. Appearance of fractures, silky, lamellar.

g. Fractured Plate B along outside row of rivet holes; tore apart from one edge. Fractures also started in Plate A at opposite edge. Appearance of fractures, silky, lamellar.

h. Fractured both plates along outside row of rivet holes. The separation of Plate A was complete; Plate B fractured through four sections. Appearance of fractures, silky, slightly lamellar.

i. Fractured both plates along outside row of rivet holes. Plate B did not separate at one edge. Appearance of fractures, silky, slightly lamellar; metal well drawn down.

j. Fractured Plate B, taking zigzag course through two outside rows of rivet holes. Appearance of fractures, silky in part, granular in part; the metal in the silky sections well drawn down, the granular sections not much reduced in thickness, the extremes of thickness after fracture being .665″ in the silky and .840″ in the granular metal. A loud report accompanied the fracture of the granular metal.

k. Fractured Plate A, taking a zigzag course through two outer rows of rivet holes. Appearance of fracture, silky, slightly lamellar.

l. Fractured Plate B, taking a zigzag course through two outside rows of rivet holes. Fracture silky, slightly lamellar.

m. Fractured Plate A along outside row of rivet holes. Fracture silky, slightly lamellar. Mean thickness at fracture, .56 inch.

n. Fractured Plate A along outside row of rivet holes. Fracture silky, slightly lamellar.

o. Fractured Plate A along outside row of rivet holes. Fracture silky, lamellar. One seam in fractured surface ¾″ wide.

p. Fractured Plate A along outside row of rivet holes. Fracture, silky lamellar.

q. Fracture in same place as H_1. Appearance, silky, slightly lamellar.

r. Fractured Plate B along outside row of rivet holes. Appearance, silky, slightly lamellar. Plates open at butt joint ½ inch.

s. Fractured Plate B along outside row of rivets, beginning the fractures at edges and extending from rivet holes toward middle of plate. Fracture silky, slightly lamellar; metal well drawn down.

t. Fractured Plate A along outside row of rivets. Appearance of fracture, silky, slight lamination.

u. Fractured Plate A along outside row of rivets. Appearance of fracture, silky, slightly lamellar; metal well drawn down.

v. Fractured Plate B; followed outside row of rivet holes in part, and thence, through inside rows, to end of plate; sheared two end rivets. Fractures silky, slightly lamellar.

w. Fractured Plate A along outside row of rivet holes, except end sections and one middle section. Appearance, silky, lamellar.

x. Sheared every rivet in the joint in both plates. The under butt strap dropped to the floor. Double shear in Plate A, three rivets; single shear in Plate B, with the exception of one rivet, which sheared in two planes.

TENSILE TESTS OF STRIPS FROM FRACTURED ENDS OF RIVETED JOINTS.

Joint Test Number.	Marks on Piece.	Contraction in 10 Inches from — Bright red.	Contraction in 10 Inches from — Black heat.	Dimensions. Width.	Dimensions. Thickness.	Sectional Area.	Elastic limit (approximate). Total.	Elastic limit (approximate). Per sq. inch.	Tensile Strength. Total.	Tensile Strength. Per sq. inch.	Load at the Time of Fracture. Total.	Load at the Time of Fracture. Per sq. Inch.
		Inch.	*Inch.*	*Inches*	*Inches*	*Sq. inches.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*
911	A_2	.104	.081	1.460	.647	.944	31,800	33,690	51,880	54,960	45,800	96,020
		Not annealed.		1.486	.647	.961	34,800	36,210	55,600	57,860	44,900	97,610
913	B_1	.111	.084	1.448	.630	.912	30,600	33,550	52,220	57,260	45,000	97,400
		Not annealed.		1.510	.628	.948	38,000	40,080	56,400	59,490	45,500	89,220
916	D_1	.108	.086	1.440	.844	1.215	37,400	30,780	62,750	51,650	48,000	103,000
		Not annealed.		1.472	.845	1.244	39,500	31,750	66,600	53,540	50,500	95,280
919	E_1	.121	.079	1.473	.850	1.252	41,000	32,750	69,330	55,370	55,500	102,780
		Not annealed.		1.457	.850	1.238	42,500	34,330	70,600	57,030	59,000	91,610
922	G_1	.126	.078	1.460	.847	1.237	40,300	32,580	63,350	51,210	46,000	106,480
		Not annealed.		1.393	.840	1.170	Not well defined.		62,550	53,460	47,800	95,800
925	H_1	.117	.080	1.474	.615	.907	32,700	36,050	52,080	57,420	43,000	101,900
		Not annealed.		1.497	.611	.915	35,000	38,250	54,250	59,290	44,500	98,020
931	K_1	.108	.076	1.450	.610	.885	31,100	35,140	51,040	57,670	41,200	101,970
		Not annealed.		1.408	.605	.852	32,500	38,150	50,800	59,630	41,000	95,790

Joint Test Number.	Mark on Piece.	Condition.	Elongation in 10 Inches.		Area of Fracture.	Contraction of Area.	Appearance of Fracture.	Elongation of Inch Sections.
			Inches.	*Per Ct.*	*Inches.* *sq. ins.*	*P. c.*		*Inches.*
911	A_2	Annealed.	2.52	25.2	1.06 × .45 = .477	49.5	Silky, lamellar.	**.62**, .23, .21, .21, .22, .21, .21, .23, .21, .17
		Not anld.	1.13	11.3	1.07 × .43 = .460	52.1	" "	.37, .17, .12, .09, .08, .06, .07, .05, .06, .06
913	B_1	Annealed.	2.40	24.0	1.05 × .44 = .462	49.3	Silky, stratified.	.15, .14, .14, .17, .19, .24, **.50**, .46, .23, .18
		Not anld.	1.33	13.3	1.16 × .44 = .510	46.2	" "	**.67**, .20, .11, .08, .06, .05, .04, .05, .03, .04
916	D_1	Annealed.	2.84	28.4	.95 × .49 = .466	61.6	Silky, slightly lamellar.	.21, .21, .20, .25, .43, **.73**, .27, .22, .16, .16
		Not anld.	.82	8.2	1.04 × .51 = .530	57.4	Silky......................	.06, .03, .04, .05, .05, .05, .09, .07, .14, .24
919	E_1	Annealed.	2.97	29.7	1.00 × .54 = .540	56.9	Silky, slightly lamellar.	.19, .17, .21, .25, .30, .33, **.79**, .29, .25, .19
		Not anld.	.67	6.7	1.11 × .58 = .644	48.0	Silky......................	.04, .04, .02, .03, .05, .05, .05, .09, .11, .19
922	G_1	Annealed.	3.16	31.6	.90 × .48 = .432	65.1	"	.16, .16, .19, .22, .28, .31, .32, **.90**, .34, .28
		Not anld.	.65	6.5	.96 × .52 = .499	57.4	"	.29, .09, .08, .06, .04, .03, .02, .02, .01, .0[illegible]
925	H_1	Annealed.	2.53	25.3	1.03 × .41 = .422	53.5	Silky, lamellar............	.23, .27, **.68**, .34, .24, .18, .16, .16, .12, .15
		Not anld.	.87	8.7	1.08 × .42 = .454	50.4	Silky, slightly lamellar.	.36, .15, .12, .09, .05, .03, .02, .03, .01, .01
931	K_1	Annealed.	2.32	23.2	1.01 × .40 = .404	54.4	" " "	**.71**, .27, .20, .20, .20, .17, .17, .16, .13, .11
		Not anld.	1.42	14.2	1.02 × .42 = .428	49.7	Silky, stratified..........	.04, .03, .04, .06, .03, .07, .11, .32, **.51**, .18

TENSION TESTS OF STRIPS CUT FROM FRACTURED ENDS OF RIVETED JOINTS.

These strips were taken from the middle of the width of the joint plates and parallel to the direction the joints were pulled.

Two strips were taken from each; one was annealed by heating bright red and cooling in the open air; the duplicates were not annealed and were tested as taken from the fractured joint.

When the annealed specimens were at the maximum temperature, centre-punch marks, 10″ apart, were stamped on one edge, and about the time the color had left them they were marked again on the other edge. After cooling to 70° Fahr. the distances between these marks were measured.

The amount of contraction, therefore, indicates approximately the heat at which the strips were annealed.

48. Riveting Machines.

Rivets are driven either by a succession of relatively light blows, as in hand-work and by pneumatic hammers, or by heavy and sustained pressure, as in hydraulic machines. In the latter process, the continuous and powerful compression of the hot rivet-blank upsets the shank, fills the hole, and closes the plates, while in hammering, especially if the blows are light, the head may be formed before the shank is upset fully, the rivet may be more or less loose in its hole, and the impact tends to crystallize the metal. Owing, however, to the extremely rapid action of the pneumatic hammer, excellent results in hull and structural work have been obtained by its use. When, as in marine cylindrical boilers, the plates are thick and the rivets large, hydraulic riveting is necessary in order to secure tight joints.

Riveting machines may be "fixed" and powerful, as for steam boilers and shop-riveting in general, or light and portable, as in the types used for hull work and field-rivets. The essential parts of any machine are a stationary "stake" holding the die which engages the rivet-head and a piston or ram driving a second die which upsets the shank and forms the point. The stake or its equivalent forms part of the framing of the machine. In the pneumatic hammer, it is replaced by an air-pressure mechanism known as the "pneumatic holder-on." In portable riveters, the riveting plunger may be direct-acting or be operated through linkage from the piston rod. An auxiliary cylinder, actuating a plate-closing device, has been used for clamping the joint before the rivet is upset.

Either steam, hydraulic, or pneumatic power is used in riveting machines. All are applied to drive by pressure and the latter, in the pneumatic hammer, by impact as well. Steam has the advantages of familiar mechanism and the absence of an accumulator

or compressor-plant; but its relatively low pressure makes large cylinders necessary, and, owing to its condensation, expansibility, and the leakage inevitable with piston-valves, the pressure developed is not uniform and is delivered largely in the form of a blow which tends to crystallize the rivet. These objections, except with regard to condensation, apply in the main to pneumatic machines, although their portability gives them a wide field. In hydraulic riveting, a pressure of 1,500 lbs. per sq. in. is used. This gives a small cylinder requiring, relatively, but little fluid, while the practically incompressible and inexpansible character of the latter makes the driving stroke a powerful and uniform squeezing of the metal which fills the hole and forms the point without impact. The high pressure, however, necessitates strong and accurately made joints, and, with careless handling in winter weather, waste water in cylinders or pipes may freeze. Since

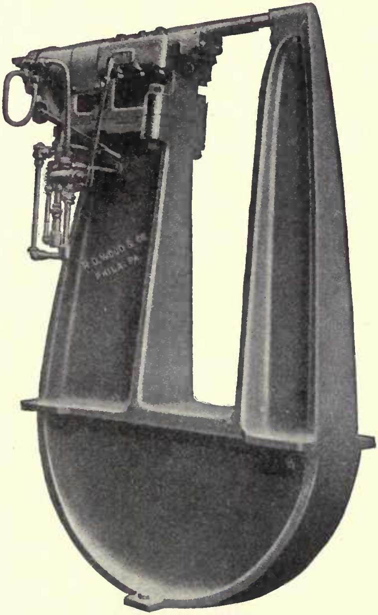


FIG. 83.

the fluid in the driving line is under 1,500 lbs. pressure, its liability to freezing is slight. A mixture of one third crude glycerine and two thirds water is used with success in Canada and northern Russia.

Riveting machines, in their power, form, and fixed or portable character, present a wide variety of types. The descriptions given below refer to the two which may be considered as the extremes of this range.

1. HYDRAULIC FIXED RIVETER.—Fig. 83 shows in elevation the riveter of this type built by Messrs. R. D. Wood and Company.

It is of "triple power," *i. e.*, it exerts any one of three pressures upon the rivet, thus fitting it not only for work of the heaviest character, but also for that upon light plates which would be crushed by the pressures required for rivets of large diameter. The powers and sizes of the standard machines of this type are :

No. 1— 50,	35 or 15 tons power	5′, 6′, 7′,	8′,	9′6″,	10′6″	and	12′	gaps.
" 2— 60,	40 " 20 " "	5′, 6′, 7′,	8′,	9′6″,	10′6″	"	12′	"
" 3— 75,	50 " 25 " "	7′, 8′,	10′6″,		12′	"	17′	"
" 4—100,	67 " 33 " "	8′,	10′6″,		12′	"	17′	"
" 5—150,	100 " 50 " "	8′, 9′6″,	10′6″,		12′	"	17′	"
" 6—180,	120 " 60 " "	8′, 9′6″,	10′6″,		12′	"	17′	"

Usual working pressure, 1,500 pounds per square inch.

The frame is a single casting to which the cylinder is bolted, the joint being tongued and grooved to ensure absolute rigidity. The cylinder, glands, rams, the framing, and hence the stakes, are made from open-hearth steel castings, having an ultimate tensile strength of 70,000 lbs. per sq. in., an elastic limit of 40,000 lbs. per sq. in., and an elongation of 20 per cent. in an 8-in. test-piece.

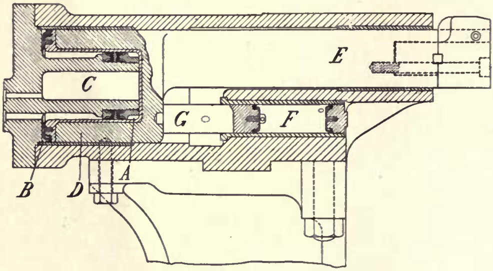


FIG. 84.

Fig. 84 gives a vertical section through a riveting head, with rams having inside packing of leather. The section in Fig. 85 is similar, excepting that the rams are packed outside with flax. In the latter arrangement, the packing in the three stuffing boxes is held in place by outside glands and is, hence, accessible readily for repacking or adjustment.

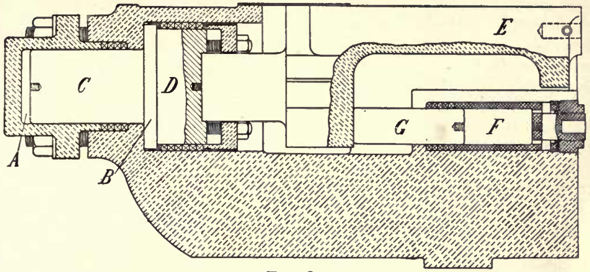


FIG. 85.

The operation of the ram is the same in each case. Referring to Fig. 85, it will be seen that there are tandem cylinders, *A* and *B*, in which the duplex ram, *CD*, carrying the riveting head, *E*, reciprocates. In the additional cylinder, *F*, the "pull-back ram," *G*, moves. The riveter is fitted with a distributing valve and an operating valve. The former is practically a double-stop valve and may be adjusted in any one of three positions, viz.: With the water passage to the small cylinder, *A*, open and that to the large cylinder, *B*, closed; with the passage open to *B* and closed to *A*; with the passages open to both cylinders. The ram-areas upon which the accumulator-pressure acts are: With the first adjustment, that of cylinder, *A*; with the second, that of the difference in area between cylinders *B* and *A*; and, with the third, the full area of cylinder, *B*. The operating valve is of the balanced piston type with leather packing. The accumulator-pressure is led directly to the pull-back ram without passing through either of the valves as above. This ram is, hence, always in action and its back pressure must be overcome by that in the driving cylinders before the ram, *CD*, can advance. In riveting, the operator first sets the distributing valve for the pressure desired; then moves the plates until the rivet is opposite the dies and throws the operating lever. Until the latter is withdrawn to its original position, the pressure remains upon the rivet. Since the operation of the type shown in Fig. 84 is the same as that just described, the reference-letters for similar parts in both are identical.

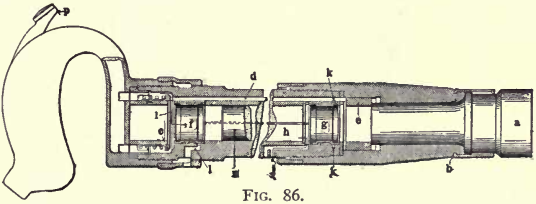


FIG. 86.

2. PNEUMATIC RIVETING HAMMER. — Fig. 86 * gives a longitudinal section of the Boyer "Long-Stroke Pneumatic Hammer."

Compressed air is admitted to the hammer through a hose coupled at the lower extremity of the handle. The admission is controlled by a main throttle valve of the

* *American Machinist*, April 25, 1901.

balanced piston type. This valve is closed by a spring and is depressed and opened by the throttle-lever, *p*, which is pressed by the thumb of the operator. The riveting die, *a*, is held in position by the light clip, *b*, only. Hence, if it is not pressed against the rivet, the first blow of the hammer, *r*, would discharge it like a bullet. To prevent this, an auxiliary spring-pressed throttle or stop-valve, *c*, is fitted, which valve is operated by two rods, as *d*, which extend through the body of the hammer and have their outer ends resting upon the ring *e*, against which the die-shank abuts. When, therefore, the die is not forced firmly against a rivet — although the main throttle-valve may be open — the auxiliary valve *c* will be seated by air and spring pressure and the rods *d*, ring *e*, and die *a*, will be moved slightly to the right. Since both valves must be open before the piston, *r*, will act, the auxiliary valve, *c*, forms a safeguard.

The valves which control the air in its pasage to and from the ends of the cylinder are shown at *f* and *g*. They are hollow and of short stroke. Rods, as *h*, similar to *d*, lie between the valves in the walls of the cylinder. As one valve moves toward the centre of the cylinder to admit air at its end, it, through the rods, pushes the other valve away from the centre, so that the exhaust is open at that end.

In the position shown, the air enters the inner end of the cylinder, as indicated by the arrow *i*, and drives the piston, *r*, outward, the exhaust escaping as shown at *j*. When the outer end of the piston enters the valve, *g*, it compresses the air before it, forming a cushion which, acting upon the annular end of *g*, pushes the latter to the left and hence valve *f* also through the rods, *h*. In this position, the ports which were open previously are closed, port *k* is open to live air, and exhaust occurs through port *l*. The holes in which the rods, *d*, lie, serve also as passages for air to the port *k*. On the completion of the return stroke, the piston enters the inner valve, *f*, and the valves are driven to the right.

FIG. 87.

This hammer weighs about 17 lbs. and is made for driving rivets from $\frac{5}{8}$ in. to $1\frac{1}{4}$ in., diameter. It is stated that it requires 20 cubic feet of free air per minute. The air-pressure which determines the force of the blow, ranges from 90 to 100 lbs. per sq. in., The hammer may be used without a supporting frame.

Its operation, with such frame and with the pneumatic holder-on, is illustrated by Fig. 87.

49. Riveted Joints, Marine Boilers.

Marine cylindrical boilers are of the internally fired type. Fig. 88 shows one half of a longitudinal and one half of a transverse section of the double-ended boilers of the U. S. Battleship *Kearsarge*. The diameter is 15 ft., 8 in.; the length, 21 ft. As

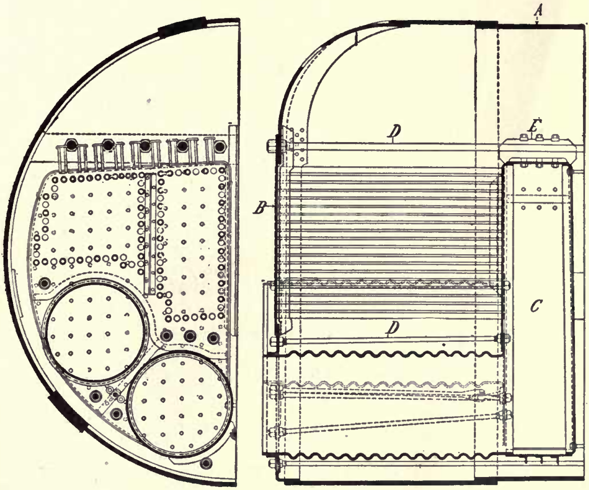


FIG. 88.

shown, the shell, *A*, is made of three courses of four plates each; the front and back heads, *B*, are each built up of three plates, the upper of which is curved backward to meet the shell; the furnaces are cylindrical and are corrugated to give strength; the combustion chamber, *C* (one to each pair of furnaces and two to each end), is built of flat plates throughout, except at the outer side which is curved so that it is concentric with the shell. The boiler is braced by stays, *D*, in the steam and water spaces, girders, *E*, upon the tops of the combustion chambers, screw-stays at the sides and backs of the latter, and by stay-tubes.

The longitudinal seams of the shell are double-strapped butt-joints, treble-riveted; the circumferential seams (central) are lap-joints, treble-riveted; the joints of heads with shell are lapped and double-riveted, except with the curved plates which are treble-riveted; the head plates are united by lapped seams — the upper, quadruple, the lower, double-riveted; all joints in furnaces and combustion chambers are single-riveted lap seams.

1. Rivet and Plate Metals. — The physical and chemical characteristics of rivet-metals, as prescribed in the specifications (1901) of the Bureau of Steam Engineering, U. S. Navy, have been given in § 32. The tests for rivets, as laid down in these specifications, are:

Rivets. — Samples from each lot are to stand the following tests without fracture, test (*a*) being applied to one lot, and (*b*) to a second, etc.:

(*a*) Bend double cold to a curve of which the inner diameter is equal to the diameter of the rivet.

(*b*) Bend double hot through an angle of 180° flat back.

(*c*) The head to be flattened when hot without cracking at the edges until its diameter is two and one half times the diameter of the shank.

(*a*) The shanks of sample rivets to be nicked on one side and bent cold to show the quality of the material.

Surface Inspection. — Rivets shall be true to form, concentric, and free from injurious scale, fins, seams, and all other injurious defects. If the material is found to be very uniform and none of the tests made of a series of lots fail, the inspector may discontinue the tests after he has made enough to satisfy himself that the whole of the material on the order is satisfactory.

Note. — In measuring the diameter of rivets the inspector will allow for the trade custom of making rivets with an actual diameter slightly (about $\frac{1}{16}$ of an inch) less than the nominal diameter.

Class.	Material.	Minimum Tensile Strength.	Minimum Elastic Limit.	Elongation.	Maximum Amount of.		Cold Bend about an Inner Diameter.
					P.	S.	
		Lbs. per sq. in.	*Lbs. per. sq. in.*	*Per ct. in 8 inches.*			
Class A.	Open-hearth steel.	70,000	37,000	22	.04	.03	Equal to thickness of plate and through 180°.
Class B.	Open-hearth steel.	60,000	32,000	25	.04	.03	Flat back through 180°.
Class C.*	Open-hearth or Bessemer.	To be in accordance with the "Standard Specifications of the Association of American Steel Manufacturers for Structural Steel," revised July, 1896.					

* Class C plates, shapes, etc., will be inspected at the building yard and not at the place of manufacture except upon special request of the contractor. No physical or chemical test will be made unless from the appearance of the plates giving evidence of overheating, cold-rolling, etc., or for other reasons, the inspector has doubts as to their fitness for the purpose for which they are intended.

The physical and chemical characteristics of steel boiler-plate, as similarly prescribed, are:

1. The physical and chemical characteristics of steel boiler-plate are to be in accordance with the table on page 206.

2. *Kind of Material.* — Steel for boiler-plates of all grades (except Class C) shall be made by the open-hearth process, and shall contain not more than four one-hundredths of 1 per cent. of phosphorus, and not more than three one-hundredths of 1 per cent. of sulphur.

2. PROPORTIONS OF RIVETS. — The standard boiler rivet for the U. S. Navy is of the "pan-head," or conical frustum, type. The head and point are alike. Table XLII. gives the proportions. In this table, a is the diameter of the rivet, b the greatest and d the least diameter of the head, and c is the height of the latter. The angle of the sides is about 65° in the 1-in. rivet. a and d are equal.

TABLE XLII.

(BOILER RIVETS, U. S. NAVY.)

a	b	c	d	Wt. of 10-Heads.
$\frac{7}{16}''$	$\frac{13}{16}''$	$\frac{3}{8}''$	$\frac{7}{16}''$	.331 lbs.
$\frac{1}{2}$	$\frac{15}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	.531 "
$\frac{9}{16}$	1	$\frac{1}{2}$	$\frac{9}{16}$	.713 "
$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	1.007 "
$\frac{11}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	1.373 "
$\frac{3}{4}$	$1\frac{5}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	1.551 "
$\frac{13}{16}$	$1\frac{7}{16}$	$\frac{11}{16}$	$\frac{13}{16}$	2.032 "
$\frac{7}{8}$	$1\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{8}$	2.258 "
$\frac{15}{16}$	$1\frac{5}{8}$	$\frac{3}{4}$	$\frac{15}{16}$	2.871 "
1	$1\frac{3}{4}$	$\frac{13}{16}$	1	3.584 "
$1\frac{1}{16}$	$1\frac{13}{16}$	$\frac{13}{16}$	$1\frac{1}{16}$	3.91 "
$1\frac{1}{8}$	$1\frac{15}{16}$	$\frac{7}{8}$	$1\frac{1}{8}$	4.761 "
$1\frac{3}{16}$	2	$\frac{7}{8}$	$1\frac{3}{16}$	5.17 "
$1\frac{1}{4}$	$2\frac{1}{8}$	$\frac{15}{16}$	$1\frac{1}{4}$	6.215 "
$1\frac{5}{16}$	$2\frac{1}{4}$	1	$1\frac{5}{16}$	7.391 "

The rivet-heads used for boilers at the Union Iron Works are "button-head" or spherical. The proportions are:

Diameter of shank $= d$;
" head $= \frac{3}{2}d + \frac{1}{16}''$;
Depth " $= \frac{5}{8}d$.

3. PROPORTIONS OF JOINTS. — The following tables give the proportions of the principal seams of typical cylindrical boilers of the U. S. Navy. The plates and rivets are of steel. d is the diameter of the rivet-hole, p is the greatest pitch, V is the

distance between the rivet-rows in staggered riveting, and V_1 the similar distance between the outer and the next row, when alternate rivets in the outer row are omitted. The general dimensions and thickness of sheets are :

Boiler No.	Working Pressure.	Diameter.	Length.	Thickness of Sheets.								
				Shell.	Head.			Furnace.	Table Sheet.	Combustion Chamber.	Butt Straps.	
					Upper.	Mid.	Lower.				Outer.	Inner.
1	180	15′ 8″	20′ 10″	$1\frac{7}{16}$	$1\frac{15}{64}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{3}{4}$	$\frac{9}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$
2	180	10 6	10 6	1	$\frac{7}{8}$	$\frac{3}{4}$	—	$\frac{17}{32}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{7}{8}$	$\frac{7}{8}$
3	180	7 $9\frac{3}{8}$	9 $9\frac{3}{8}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{5}{8}$	—	$\frac{19}{32}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{11}{16}$

The proportions of the joints are :

Boiler No.	Seam.	Kind of Joint.	d	p	V	V_1	E
1	Shell, longitudinal.	Double strapped, butt, triple riveted, zigzag, alternate rivets, outer row, omitted.	$1\frac{3}{8}$	$8\frac{5}{16}$	$2\frac{3}{8}$	$3\frac{5}{16}$	$2\frac{1}{16}$
2	" "	Double strapped, butt, triple riveted, zigzag, alternate rivets, outer row, omitted.	$1\frac{3}{16}$	7	$1\frac{7}{8}$	$2\frac{11}{16}$	$1\frac{9}{16}$
3	" "	Double strapped, butt, triple riveted, zigzag, alternate rivets, outer row, omitted.	$\frac{7}{8}$	$5\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{4}$
1	Shell, circumferential.	Lap, triple riveted, zigzag.	$1\frac{1}{2}$	$4\frac{7}{32}$	$2\frac{3}{8}$		$2\frac{1}{4}$
2	" "	Lap, double riveted, zigzag.	$1\frac{3}{8}$	$3\frac{5}{8}$	$2\frac{1}{8}$		$2\frac{1}{16}$
3	" "	" " " "	$1\frac{3}{16}$	$3\frac{7}{16}$	$1\frac{7}{8}$		$1\frac{25}{32}$
1	Head to shell.	" " " "	$1\frac{5}{16}$	$4\frac{1}{4}$	$2\frac{1}{4}$		$1\frac{7}{8}$
2	" "	" " " "	$1\frac{1}{8}$	$3\frac{5}{16}$	$1\frac{3}{4}$		$1\frac{11}{16}$
3	" "	" " " "	$1\frac{1}{16}$	$3\frac{1}{16}$	$1\frac{5}{8}$		$1\frac{1}{2}$
1	Front head, upper.	Lap, quadruple riveted, zigzag. V_1 lower row $= 2\frac{7}{16}$.	$1\frac{5}{16}$	5	$2\frac{3}{4}$		$1\frac{7}{8}$
2	" "	Lap, triple riveted, zigzag; alternate rivets, middle row omitted; pitch, outer rows, $3\frac{1}{4}$, inner, $7\frac{1}{4}$.	$1\frac{1}{16}$		$3\frac{3}{4}$		$1\frac{3}{4}$
3	" "	Lap, triple riveted, zigzag.	$1\frac{1}{16}$	$3\frac{1}{4}$	$1\frac{5}{8}$		$1\frac{1}{2}$
1	Front head, lower.	Lap, double riveted, zigzag.	1	3	$1\frac{9}{16}$		$1\frac{7}{16}$
1	Furnace to tube-sheet.	Lap, single riveted.	$\frac{15}{16}$	$2\frac{1}{4}$			$1\frac{5}{16}$
2	" "	" " "	$\frac{15}{16}$	2			$1\frac{1}{4}$
3	" "	" " "	$\frac{15}{16}$	$2\frac{1}{6}$			$1\frac{3}{8}$
1	Tube-sheet to combustion chamber and combustion chamber seams.	" " "	$\frac{15}{16}$	$2\frac{1}{4}$			$1\frac{5}{16}$
2	Tube-sheet to combustion chamber and combustion chamber seams.	" " "	1	$2\frac{3}{16}$			$1\frac{1}{4}$
3	Tube-sheet to combustion chamber and combustion chamber seams.	" " "	$\frac{15}{16}$	$2\frac{1}{8}$			$1\frac{3}{8}$

4. WEIGHT OF RIVETS. — The total weight and the weight of rivets are given below for three cylindrical boilers of large size for the U. S. Navy. The weight given is that of the boiler simply without fittings, such as grate-bars, valves, lagging, etc. The plates and rivets are of steel.

Working Pressure, lbs.	Diameter.	Length.	Total Weight of Boiler, lbs.	Weight of Rivets, lbs.	Rivet-percentage of Total Weight.
160	15′0″	18′0″	135,793	5,788	4.26
160	15′3″	21′3″	149,634	6,218	4.16
135	14′8″	19′2″	108,128	5,391	4.98

5. THE U. S. BOARD OF SUPERVISING INSPECTORS OF STEAM VESSELS. — The regulations (Jan., 1901) of this board give the following formulæ for the proportions of single- and double-riveted lap-joints for both iron and steel boilers. Let:

$p =$ greatest pitch of rivets, ins.;
$n =$ number of rivets in one pitch-section;
$p_d =$ diagonal pitch, ins.;
$d =$ diameter of rivets, ins.;
$T =$ thickness of plate, ins.;
$V =$ distance between rows of rivets, ins.;
$E =$ distance from edge of plate to centre of rivet, ins.

For *iron* plates and *iron* rivets:

$$p = \frac{d^2 \times .7854 \times n}{T} + d.$$

For *steel* plates and *steel* rivets:

$$p = \frac{23 \times d^2 \times .7854 \times n}{28 \times T} + d.$$

For all joints:

$$E = \frac{3}{2} d.$$

For double *chain*-riveted joints, V should not be less than $2d$; but it is more desirable that V should not be less than $\frac{4d+1}{2}$. For ordinary, double, *zigzag*-riveted joints:

$$V = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10}.$$

For double, zigzag-riveted lap joint, iron and steel:

$$p_d = \frac{6p + 4d}{10}.$$

For single-riveted lap joints:

$$\textit{Maximum pitch} = (1.31 \times T) + 1\tfrac{5}{8}.$$

For double-riveted lap joints:

$$\textit{Maximum pitch} = (2.62 \times T) + 1\tfrac{5}{8}.$$

These formulæ are equivalent to those of the British Board of Trade and are similar in many respects to those given in Traill's handbook (§ 43). From the latter, tables of single- and double-riveted lap joints, for both iron and steel, are quoted in, and authorized for use by, these regulations.

6. PROCESS OF RIVETING. — U. S. Naval specifications for boilers require that "hydraulic riveting shall be used wherever possible. In parts where hydraulic riveting cannot be used, the rivet-holes shall be coned and conical rivets used. Seams will be calked on both sides in an approved manner."

50. Riveted Joints, Locomotive Boilers.

The following data refer to the practice of the Baldwin Locomotive Works.

1. RIVET AND PLATE METALS. — The specifications are:

Boiler and Fire-Box Steel. — All plates must be rolled from steel manufactured by the open-hearth process, and must conform to the following chemical analysis:

		Boiler Steel.	Furnace Steel.
Carbon, between		0.15 and 0.25 per cent.	1.15 and 0.25 per cent
Phosphorus, not over		0.05 per cent.	0.03 per cent.
Manganese,	"	0.45 "	0.45 "
Silicon,	"	0.03 "	0.03 "
Sulphur,	"	0.05 "	0.035 "

No sheets will be accepted that show mechanical defects. A test strip taken lengthwise from each sheet rolled and without annealing should have a tensile strength of 60,000 pounds per square inch, and an elongation of 25 per cent. in section originally 8 inches long. Sheets will not be accepted if the test shows a tensile strength of less than 55,000 pounds, or greater than 65,000 pounds per square inch, nor if the elongation falls below 20 per cent.

Fire-Box Copper. — Copper plates for fire-boxes must be rolled from best quality Lake Superior ingots; they must have a tensile strength of not less than 30,000 pounds per square inch, and an elongation of at least 20 per cent. in section originally 2 inches long.

Stay-Bolt Iron. — Iron for stay-bolts must be double-refined, and show an ultimate tensile strength of at least 48,000 pounds per square inch, with a minimum elongation of 25 per cent. in a test section 8 inches long. Pieces 24 inches long must stand bending double, both ways, without showing fracture or flaw. Iron must be rolled true to gauges furnished, and permit of cutting a clean, sharp thread.

Copper Stay-Bolts. — Copper stay-bolts must be manufactured from the best Lake Superior ingots; they must have an ultimate tensile strength of not less than 30,000 pounds per square inch, and an elongation of at least 20 per cent. in section originally 2 inches long.

The general practice of this company is to use iron rivets of the quality required as above for stay-bolts.

2. Process of Riveting. — All parts of the boiler which can be reached by fixed or portable machines are riveted by hydraulic pressure. The latter for iron or steel rivets is:

$1\frac{1}{4}$	in. diameter,	100	tons.
$1\frac{1}{8}$	"	75	"
1	"	66	"
$\frac{7}{8}$	"	50	"
$\frac{3}{4}$	"	33	"
$\frac{5}{8}$	"	25	"

For copper rivets, a pressure ranging from 25 to 33 tons — never exceeding the latter — is used. The driving head of the rivet is made the same in height as the diameter of the shank.

3. Proportions of Joints. — Tables XLIII., XLIV., XLV., XLVI. give the size and arrangement of rivets for various thicknesses of sheets in single- and double-riveted lap seams and quadruple- and sextuple-riveted butt joints, with double straps unequal in width.

Table XLIII.

Single-Riveted Longitudinal Seams. (For all Pressures.) For Outside Fire-Box Seams of Radial Stay Boilers.
(Baldwin Locomotive Works.)

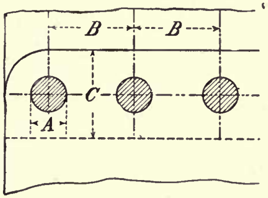


Plate.		A	B	C
Thickness.	Material.			
$\frac{1}{2}''$	Iron.	$\frac{7}{8}''$	$2\frac{1}{4}''$	$2\frac{5}{8}''$
$\frac{7}{16}$	Steel.	$\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{5}{8}$
$\frac{7}{16}$	Iron.	$\frac{3}{4}$	2	$2\frac{1}{4}$
$\frac{3}{8}$	Steel.	$\frac{3}{4}$	2	$2\frac{1}{4}$
$\frac{3}{8}$	Iron.	$\frac{3}{4}$	2	$2\frac{1}{4}$
$\frac{5}{16}$	Steel.	$\frac{3}{4}$	2	$2\frac{1}{4}$
$\frac{5}{16}$	Iron.	$\frac{5}{8}$	$1\frac{3}{4}$	2
$\frac{1}{4}$	Steel.	$\frac{5}{8}$	$1\frac{3}{4}$	2
$\frac{1}{4}$	Iron.	$\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
$\frac{1}{2}$	Steel.	$\frac{7}{8}$	2	$2\frac{5}{8}$
$\frac{9}{16}$	"	1	$2\frac{1}{4}$	3
$\frac{1}{2}$	"	1	$2\frac{1}{2}$	3
$\frac{5}{8}$	"	1	$2\frac{1}{4}$	3
$\frac{11}{16}$	"	$1\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{1}{2}$

TABLE XLIV.

DOUBLE-RIVETED SEAMS.
(BALDWIN LOCOMOTIVE WORKS.)

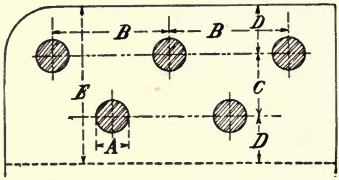


Plate. Thickness.	Plate. Material.	A	B	C	D	E	Per cent. of solid plate.
$\frac{3}{8}''$	Steel.	$\frac{3}{4}''$	$2''$	$1\frac{1}{2}''$	$1\frac{1}{8}''$	$3\frac{3}{4}''$	62
$\frac{7}{16}$	"	$\frac{7}{8}$	$2\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{5}{16}$	$4\frac{1}{4}$	65
$\frac{1}{2}$	"	$\frac{7}{8}$	$2\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{5}{16}$	$4\frac{1}{4}$	65
$\frac{9}{16}$	"	1	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$4\frac{3}{4}$	63
$\frac{5}{8}$	"	1	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$4\frac{3}{4}$	63
$\frac{11}{16}$	"	$1\frac{1}{8}$	3	$1\frac{7}{8}$	$1\frac{11}{16}$	$5\frac{1}{4}$	62
$\frac{3}{4}$	"	$1\frac{1}{4}$	$3\frac{1}{2}$	2	$1\frac{3}{4}$	$5\frac{1}{2}$	64

TABLE XLV.

QUADRUPLE BUTT-JOINT SEAMS WITH WELDED ENDS.
(BALDWIN LOCOMOTIVE WORKS.)

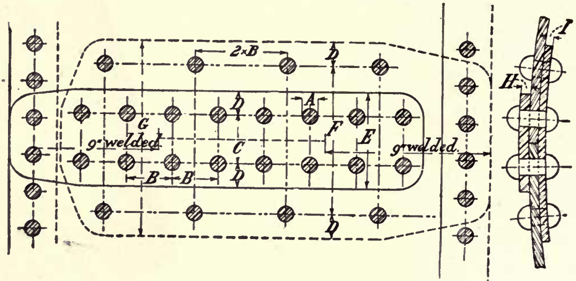


Plate. Thickness.	Plate. Material.	A	B	C	D	E	F	G	H	I	%
$\frac{3}{8}''$	Steel.	$\frac{3}{4}''$	$2''$	$2\frac{1}{4}''$	$1\frac{1}{8}''$	$4\frac{1}{2}''$	$6\frac{3}{4}''$	$9''$	$\frac{3}{8}''$	$\frac{5}{16}''$	81.2
$\frac{7}{16}$	"	$\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$1\frac{5}{16}$	$5\frac{1}{4}$	8	$10\frac{3}{4}$	$\frac{7}{16}$	$\frac{3}{8}$	82.5
$\frac{1}{2}$	"	$\frac{7}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$1\frac{5}{16}$	$5\frac{1}{4}$	8	$10\frac{3}{4}$	$\frac{1}{2}$	$\frac{7}{16}$	82.5
$\frac{9}{16}$	"	1	$2\frac{3}{4}$	3	$1\frac{1}{2}$	6	9	12	$\frac{9}{16}$	$\frac{7}{16}$	81.8
$\frac{5}{8}$	"	1	$2\frac{3}{4}$	3	$1\frac{1}{2}$	6	9	12	$\frac{5}{8}$	$\frac{7}{16}$	81.8
$\frac{11}{16}$	"	$1\frac{1}{8}$	3	$3\frac{1}{4}$	$1\frac{5}{8}$	$6\frac{1}{2}$	$9\frac{3}{4}$	13	$\frac{11}{16}$	$\frac{1}{2}$	81.2
$\frac{3}{4}$	"	$1\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	7	$10\frac{1}{2}$	14	$\frac{3}{4}$	$\frac{1}{2}$	80.7
$\frac{13}{16}$	"	$1\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	7	$10\frac{1}{2}$	14	$\frac{3}{4}$	$\frac{1}{2}$	80.7

Table XLVI.

Sextuple Butt-Joint Seams with Welded Ends.
(Baldwin Locomotive Works.)

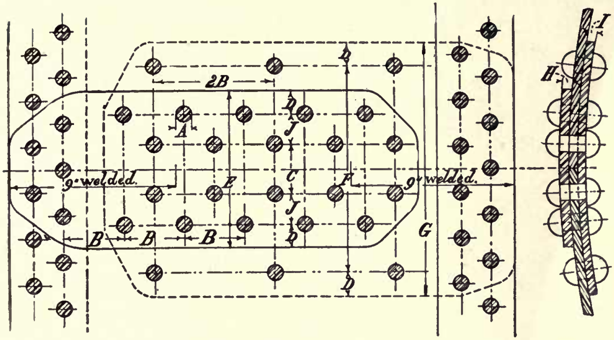


Plate.		A	B	C	D	E	F	G	H	I	J	%
Thickness.	Material.											
$\frac{3}{8}''$	Steel.	$\frac{3}{4}''$	$3''$	$2\frac{1}{4}''$	$1\frac{1}{8}''$	$7\frac{1}{2}''$	$9\frac{3}{4}''$	$12''$	$\frac{3}{8}''$	$\frac{5}{16}''$	$1\frac{1}{2}''$	87
$\frac{1}{2}$	"	$\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{5}{8}$	$1\frac{5}{16}$	$8\frac{1}{2}$	$11\frac{1}{8}$	$13\frac{3}{4}$	$\frac{1}{2}$	$\frac{7}{16}$	$1\frac{5}{8}$	86.5
$\frac{9}{16}$	"	1	$3\frac{1}{2}$	3	$1\frac{1}{2}$	$9\frac{1}{2}$	$12\frac{1}{2}$	$15\frac{1}{2}$	$\frac{9}{16}$	$\frac{7}{16}$	$1\frac{3}{4}$	85.7
$\frac{5}{8}$	"	1	$3\frac{1}{2}$	3	$1\frac{1}{2}$	$9\frac{1}{2}$	$12\frac{1}{2}$	$15\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{3}{4}$	85.7
$\frac{11}{16}$	"	$1\frac{1}{8}$	$3\frac{3}{4}$	$3\frac{1}{4}$	$1\frac{5}{8}$	$10\frac{1}{4}$	$13\frac{1}{2}$	$16\frac{3}{4}$	$\frac{11}{16}$	$\frac{1}{2}$	$1\frac{7}{8}$	85.0
$\frac{3}{4}$	"	$1\frac{1}{4}$	4	$3\frac{1}{2}$	$1\frac{3}{4}$	11	14	$17\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{2}$	2	84.0
$\frac{13}{16}$	"	$1\frac{1}{4}$	4	$3\frac{1}{2}$	$1\frac{3}{4}$	11	14	$17\frac{1}{2}$	$\frac{13}{16}$	$\frac{9}{16}$	2	84.0
$\frac{7}{8}$	"	$1\frac{1}{4}$	$4\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	11	$14\frac{1}{2}$	18	$\frac{7}{8}$	$\frac{9}{16}$	2	85.0

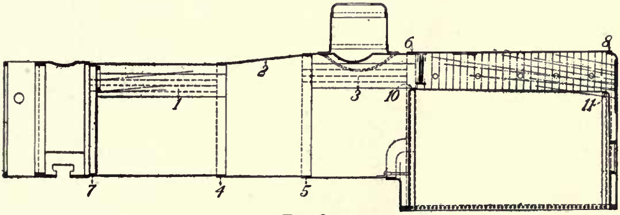


Fig. 89.

4. Location of Joints.—Fig. 89 gives a longitudinal section, omitting tubes and braces, of a Radial Stay, Wagon-top, locomo-

tive boiler, as built by this company. In Fig. 90 there is shown a transverse semi-section through the fire-box. The barrel is built of three courses of one sheet each, the thicknesses, beginning at the furnace tube-sheet, being $\frac{3}{4}$ in., $\frac{3}{4}$ in., and $\frac{11}{16}$ in., respectively. The remaining shell sheet is $\frac{9}{16}$ in. thick. The thicknesses of the tube-sheets, crown-sheet, and fire-box front and side-sheets are, respectively, $\frac{1}{2}$ in., $\frac{3}{8}$ in., and $\frac{5}{16}$ in. The longitudinal and circumferential seams of the barrel are, respectively, quadruple-riveted butt (unequal straps) and double-riveted lap joints; the remaining seams are single-riveted lap joints. Since rivet-heads in the furnace are liable to be burned off, the rivets are counter-sunk in fire-box and side sheets for 36 ins. from the bottom upward. The proportions of the principal seams are given in the following list, the notation being that of the joint-tables previously given and the locations being numbered in Figs. 89 and 90.

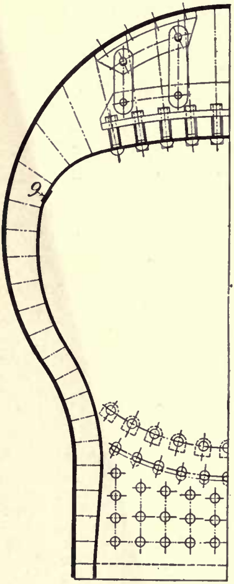


FIG. 90.

Seam. No.	Seam. Kind.	Plate. Thick.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*
1	Quadruple, butt.	$\frac{11}{16}''$	$1\frac{1}{8}''$	$3''$	$3\frac{1}{4}''$	$1\frac{5}{8}''$	$6\frac{1}{2}''$	$9\frac{3}{4}''$	$13''$	$\frac{11}{16}''$	$\frac{1}{2}''$
2	" "	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	7	$10\frac{1}{2}$	14	$\frac{3}{4}$	$\frac{1}{2}$
3	" "	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$	7	$10\frac{1}{2}$	14	$\frac{3}{4}$	$\frac{1}{2}$
4	Double-riveted lap.	$\frac{11}{16}$, $\frac{3}{4}$	$1\frac{1}{8}$	3	$1\frac{7}{8}$	$1\frac{11}{16}$	$5\frac{1}{4}$				
5	" "	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{2}$	2	$1\frac{3}{4}$	$5\frac{1}{2}$				
6	" "	$\frac{9}{16}$, $\frac{3}{4}$	1	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$4\frac{3}{4}$				
7	Single-riveted lap.	$\frac{1}{2}$, $\frac{11}{16}$	$\frac{7}{8}$	2	$2\frac{5}{8}$						
8	" "	$\frac{9}{16}$	1	$2\frac{1}{4}$	3						
9	" "	$\frac{5}{16}$, $\frac{3}{8}$	$\frac{3}{4}$	2	$2\frac{1}{4}$						
10	" "	$\frac{5}{16}$, $\frac{3}{8}$, $\frac{1}{2}$	$\frac{3}{4}$	2	$2\frac{1}{4}$						
11	" "	$\frac{5}{16}$, $\frac{3}{8}$	$\frac{3}{4}$	2	$2\frac{1}{4}$						

51. Riveted Joints, Stationary Boilers.

Cylindrical boilers for stationary service are usually of the externally fired type, the shell containing only the bracing and the tubes or flues.

1. AMERICAN BOILER MANUFACTURERS' ASSOCIATION. — The following extracts from the *Uniform American Boiler Specifications* adopted in October, 1898, by this association are given through the courtesy of E. D. Meier, Esq., chairman of the committee which formulated these specifications.

2. *Steel.* — Homogeneous steel made by the open hearth or crucible processes, and having the following qualities, is to be used in all boilers:

Tensile Strength, Elongation, Chemical Tests. — Shell plates *not* exposed to the direct heat of the fire or gases of combustion, as in the external shells of internally fired boilers, may have from 65,000 to 70,000 pounds tensile strength; elongation not less than 24 per cent. in 8 inches; phosphorus not over .035 per cent.; sulphur not over .035 per cent.

Shell plates in any way exposed to the direct heat of the fire or the gases of combustion, as in the external shells or heads of externally fired boilers, or plates on which any flanging is to be done, to have from 60,000 to 65,000 pounds tensile strength; elongation not less than 27 per cent. in 8 inches; phosphorus not over .03 per cent.; sulphur not over .025 per cent.

Fire-box plates or such as are exposed to the direct heat of the fire, or flanged on the greater portion of their periphery, to have 55,000 to 62,000 pounds tensile strength; elongation 30 per cent. in 8 inches; phosphorus not over .03 per cent.; sulphur not over .025 per cent.

For all plates the elastic limit to be at least one half the ultimate strength; percentage of manganese and carbon left to the judgment of the steel maker. * * *

3. *Rivets* to be of good charcoal iron, or of a soft, mild steel, having the same physical and chemical properties as the fire-box plates, and must test hot and cold by driving down on an anvil with the head in a die; hy nicking and bending, by bending back on themselves cold, without developing cracks or flaws. * * *

10. *Riveting.* — Holes made perfectly true and fair by clean-cutting punches or drills. Sharp edges and burrs removed by slight countersinking and burr reaming before and after sheets are joined together.

Under side of original rivet head must be flat, square and smooth. For rivets $\frac{5}{8}$ inch to $\frac{15}{16}$ inch diameter allow $1\frac{1}{2}$ diameters for length of stock to form the head, and less for larger rivets. Allow 5 per cent. more stock for driven head for button set or snap rivets. Use light regulation riveting hammers until rivet is well upset in the hole; after that snap and heavy mauls. For machine riveting more stock to be left for driven head to make it equal to original head, as fixed by experiment.

Total pressure on the die about 80 tons for $1\frac{1}{8}$-inch to $1\frac{1}{4}$-inch rivets; 65 tons for 1-inch; 57 tons for $\frac{15}{16}$-inch; 35 tons for $\frac{3}{4}$-inch rivets.

Make heads of rivets equal in strength to shanks by making head at periphery of shank of a height equal to $\frac{1}{3}$ diameter of shank and giving a slight fillet at this point.

Approximately, make rivet holes double thickness of thinnest plate; pitch three times rivet hole; pitch lines of staggered rows $\frac{1}{2}$ pitch apart; lap for single-riveting equal to pitch, for double-riveting $1\frac{1}{3}$ pitch, and $\frac{1}{2}$ pitch more for each additional row of

rivets; *exact dimensions determined* by making resistance to shear of aggregate rivet section at least 10 per cent. greater than tensile strength of net or standing metal.

11. *Rivet Holes* punched with good sharp punches and well-fitting dies in A. B. M. A. steel up to ⅝ inch thickness; in thicker plates punch and ream with a fluted reamer, or drill the holes.

12. *Drift Pin* to be used only with light hammers to pull plates into place and round up the hole, but never to enlarge or gouge holes with heavy hammers. * * *

25. *Rivet Seams* when proportioned as prescribed in Section 10 with materials tested as per Sections 2 and 3 shall have 4½ as factor of safety; when not so tested, but inspection of materials indicates good quality, a factor of safety of 5 is to be taken, and at most 55,000 lbs. tensile strength assumed for the steel plate and 40,000 lbs. shear strength for the rivets, all figured on the actual net standing metal.

2. THE HARTFORD STEAM BOILER INSPECTION AND INSURANCE CO. — The following data refer to the practice of this company. The specifications for horizontal tubular steam boilers require that the material for shell plates and heads shall be Open Hearth Fire-box Steel and best Open Hearth Flange Steel, respectively; that the longitudinal and girth seams shall be, respectively, of the butt-

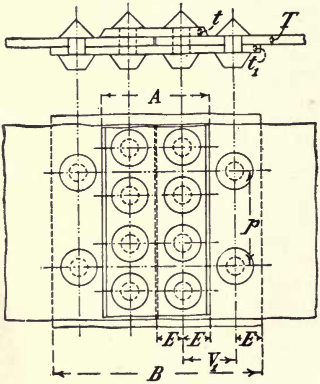


FIG. 91.

joint type with double-covering strips and the single-riveted lap-joint type; and that the rivet-holes shall be drilled in place, *i. e.*, holes punched at least $\frac{1}{4}$ in. less than full size, then courses rolled up, covering plates and heads bolted to courses with all holes together perfectly fair, rivet-holes drilled to full size, and finally plates taken apart and burrs removed. No rivets shall be driven in unfair holes; such holes must be brought in line with a reamer

Tables XLVII., XLVIII., XLIX., L. and Figs. 91 and 92 give the proportions of longitudinal and circumferential or girth seams. The inner covering strap of the butt joints is wider than the outer.

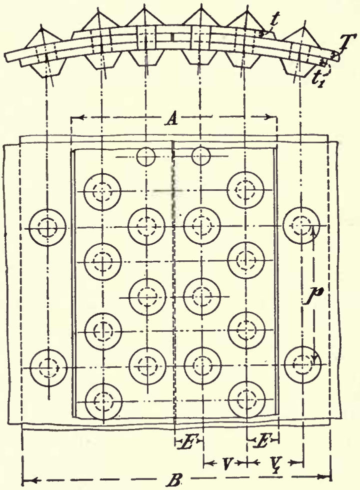


FIG. 92.

The inner row or rows of rivets have half the pitch of the outer row. The rivets of the latter pass through the plate and inner covering strap only and are thus in single shear. The joints are proportioned for steel plates and iron rivets. The tensile strength of plates is taken as 60,000 lbs. per sq. in. of section and the shearing resistance of rivets (single shear) as 38,000 lbs. per sq. in. of section. The diameter of rivet holes is $\frac{1}{16}$ in. greater than the diameter, d, of the rivets. The notation of the tables is:

$C =$ circumferential or girth seam;
$L =$ longitudinal seam;
$T =$ thickness of plate, ins.;
$t =$ thickness of outer butt-strap, ins.;
$t_1 =$ thickness of inner butt-strap, ins.;
$d =$ diameter of rivet, ins.;
$p =$ greatest pitch of rivets, ins.;

$V =$ distance between rivet-rows, staggered riveting, ins.;

$V_1 =$ distance between outer and next row, staggered riveting, when alternate rivets are omitted in outer row, ins.;

$E =$ distance from centre of nearest rivet to edge of plate or strap, ins.;

$A =$ total width of outer butt-strap, ins.;

$B =$ total width of inner butt-strap, ins.

TABLE XLVII.

CIRCUMFERENTIAL SEAM, *C*, SINGLE-RIVETED LAP; LONGITUDINAL SEAM, *L*, DOUBLE- (STAGGERED) RIVETED LAP.

(HARTFORD STEAM BOILER INSP. AND INS. CO.)

Plate Thickness.	Seam.	d	p	V	E
$\frac{1}{4}''$	C	$\frac{11}{16}''$	$2\frac{1}{16}''$		$1\frac{1}{8}''$
$\frac{1}{4}$	L	$\frac{11}{16}$	$2\frac{7}{8}$	$1\frac{15}{16}''$	$1\frac{1}{8}$
$\frac{5}{16}$	C	$\frac{3}{4}$	$2\frac{1}{8}$		$1\frac{7}{32}$
$\frac{5}{16}$	L	$\frac{3}{4}$	$2\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{7}{32}$
$\frac{3}{8}$	C	$\frac{7}{8}$	$2\frac{3}{8}$		$1\frac{13}{32}$
$\frac{3}{8}$	L	$\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{13}{32}$
$\frac{7}{16}$	C	$\frac{15}{16}$	$2\frac{7}{16}$		$1\frac{1}{2}$
$\frac{7}{16}$	L	$\frac{15}{16}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{1}{2}$
$\frac{1}{2}$	C	1	$2\frac{1}{2}$		$1\frac{19}{32}$
$\frac{1}{2}$	L	1	$3\frac{32}{100}$	$2\frac{2}{10}$	$1\frac{19}{32}$

TABLE XLVIII.

CIRCUMFERENTIAL SEAM, *C*, SINGLE-RIVETED LAP; LONGITUDINAL SEAM, *L*, TREBLE- (STAGGERED) RIVETED LAP.

(HARTFORD STEAM BOILER INSP. AND INS. CO.)

Plate Thickness.	Seam.	d	p	V	E
$\frac{1}{4}''$	C	$\frac{5}{8}''$	$2\frac{1}{16}''$		$1\frac{1}{32}$
$\frac{1}{4}$	L	$\frac{5}{8}$	3	$2''$	$1\frac{1}{32}$
$\frac{5}{16}$	C	$\frac{11}{16}$	$2\frac{1}{8}$		$1\frac{1}{8}$
$\frac{5}{16}$	L	$\frac{11}{16}$	$3\frac{1}{8}$	$2\frac{1}{16}$	$1\frac{1}{8}$
$\frac{3}{8}$	C	$\frac{3}{4}$	$2\frac{1}{8}$		$1\frac{7}{32}$
$\frac{3}{8}$	L	$\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{7}{32}$
$\frac{7}{16}$	C	$\frac{7}{8}$	$2\frac{3}{8}$		$1\frac{13}{32}$
$\frac{7}{16}$	L	$\frac{7}{8}$	$3\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{13}{32}$
$\frac{1}{2}$	C	$\frac{15}{16}$	$2\frac{1}{2}$		$1\frac{1}{2}$
$\frac{1}{2}$	L	$\frac{15}{16}$	$3\frac{15}{16}$	$2\frac{5}{8}$	$1\frac{1}{2}$

Table XLIX.

Double- (Staggered) Riveted Butt Joints with Unequal Straps. Alternate Rivets Omitted in Outer Row.

(Hartford Steam Boiler Insp. and Ins. Co.)

T	t	t_1	d	p	V_1	E	A	B	Efficiency.
$\frac{5}{16}''$	$\frac{1}{4}''$	$\frac{1}{4}''$	$\frac{11}{16}''$	$4\frac{1}{2}''$	$2\frac{1}{4}''$	$1\frac{1}{8}''$	$4\frac{1}{2}''$	$9''$	83.0 %
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{4}$	$4\frac{3}{4}$	$2\frac{7}{16}$	$1\frac{1}{4}$	5	$9\frac{7}{8}$	82.9
$\frac{7}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{13}{16}$	$4\frac{15}{16}$	$2\frac{5}{8}$	$1\frac{5}{16}$	$5\frac{1}{4}$	$10\frac{1}{2}$	82.0
$\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{8}$	$5\frac{1}{8}$	$2\frac{13}{16}$	$1\frac{13}{32}$	$5\frac{5}{8}$	$11\frac{1}{4}$	80.0

Table L.

Treble- (Staggered) Riveted Butt Joints with Unequal Straps. Alternate Rivets Omitted in Outer Row.

(Hartford Steam Boiler Insp. and Ins. Co.)

T	t	t_1	d	p	V	V_1	E	A	B	Efficiency.
$\frac{5}{16}''$	$\frac{1}{4}''$	$\frac{1}{4}''$	$\frac{11}{16}''$	$6\frac{1}{4}''$	$2\frac{1}{8}''$	$2\frac{3}{8}''$	$1\frac{1}{4}''$	$9\frac{1}{4}''$	$14''$	88.0 %
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{4}$	$6\frac{1}{2}$	$2\frac{3}{16}$	$2\frac{1}{2}$	$1\frac{7}{32}$	$9\frac{1}{4}$	$14\frac{1}{4}$	87.5
$\frac{7}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{8}$	$6\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$1\frac{13}{32}$	$10\frac{1}{8}$	$15\frac{5}{8}$	86.0
$\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{15}{16}$	$7\frac{1}{2}$	$2\frac{3}{8}$	3	$1\frac{1}{2}$	$10\frac{3}{4}$	$16\frac{3}{4}$	86.6

52. Riveted Joints, Structural Work.

The tables and other data given in this section refer principally to the practice of the American Bridge Company.

1. Rivet and Plate Metals. — General specifications for structural steel have been given in § 37. For steel railroad bridges this company requires:

All steel to be made by Open Hearth process. Per cent. of phosphorus: Acid, .08; basic, .05.

Grades.	Rivet.	Soft.	Medium.
Ult. strength, lbs. per sq. in.	48–58,000	52–62,000	60–70,000
Elongation, per cent.	26	25	22
Elastic limit.	$\frac{1}{2}$ ult. str.	$\frac{1}{2}$ ult. str.	$\frac{1}{2}$ ult. str.

For rivet and soft steel, test-piece to bend 180° flat on itself; for medium steel, 180° to a diameter equal to thickness of piece — in all cases without fracture on outside of bent portion.

In general practice, field-rivets, *i. e.*, those driven in course of erection, are frequently of wrought iron, since its range of riveting temperature is less affected than that of steel by cooling and delay.

2. Rivet Proportions. — The *diameter* ranges between $\frac{5}{8}$ in. and 1 in., the usual size being $\frac{3}{4}$ in. or $\frac{7}{8}$ in. The smaller diame-

ters are used with thin and narrow flanges and the 1 in. size only when the thickness or stress requires it. Field-rivets should not be over $\frac{3}{4}$ in., if possible. The selection of the diameter is, to some extent, a matter of judgment.

The form of the *head and point* is usually either spherical, countersunk, or flattened to $\frac{3}{8}$ in. thickness. Table LI. gives the proportions for spherical (button-head) and countersunk forms, the formulæ being:

Spherical.	Countersunk.
$D = \frac{3}{2}d + \frac{1}{8}''$,	Angle of sides $= 60°$,
$H = 0.425D$,	$H = \frac{1}{2}d$,

in which $d =$ diameter of shank, $D =$ diameter of head or point, $H =$ height of spherical or depth of countersunk head or point.

TABLE LI.

PROPORTIONS OF RIVET-HEADS.

(AMERICAN BRIDGE CO.)

Shank.	Spherical Heads.		Countersunk Heads.	
Diam.	Diam.	Height.	Diam.	Height.
$\frac{3}{8}$ in.	$\frac{11}{16}$ in.	$\frac{5}{16}$ in.	$\frac{19}{32}$ in.	$\frac{3}{16}$
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{25}{32}$	$\frac{1}{4}$
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{15}{32}$	1	$\frac{5}{16}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{17}{32}$	$1\frac{3}{16}$	$\frac{3}{8}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{5}{8}$	$1\frac{3}{8}$	$\frac{7}{16}$
1	$1\frac{5}{8}$	$\frac{11}{16}$	$1\frac{9}{16}$	$\frac{1}{2}$

The conventional *Rivet-signs* used to mark on the drawing the character of the head and point are shown in Fig. 93. To save

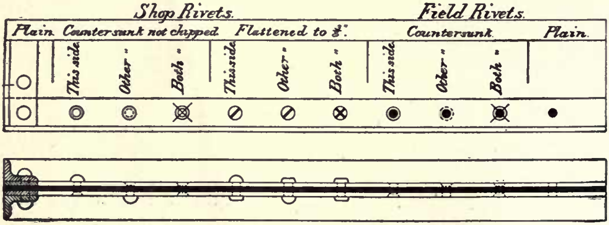


FIG. 93.

time in construction, but one size of rivet is used throughout each piece, as a plate girder, and the diameter of the rivet-holes is noted

on the drawing. The heads of countersunk rivets project usually about $\frac{1}{8}$ in. If they are required to be flush, they must be chipped.

The *length* of rivet-shank required for a given joint is equal to the "grip" plus the metal required to fill the hole and form the point. The grip is the aggregate thickness of the connected plates plus an allowance for irregularity, of $\frac{1}{32}$ in. for each place where two plate-surfaces meet. The diameter of the rivet-holes is $\frac{1}{16}$ in. greater than that of the rivets. For any given grip, the length of shank and weight of spherical (button) head steel rivets may be found from the following data:

WEIGHT IN POUNDS.

	$\frac{3}{8}''$	$\frac{1}{2}''$	$\frac{5}{8}''$	$\frac{3}{4}''$	$\frac{7}{8}''$	1″
Shank, per in. of length.	.031	.056	.087	.125	.170	.223
Two rivet-heads.	.037	.116	.222	.273	.453	.780

3. THE SPACING OF RIVETS is determined mainly by the required strength at any given point, tightness, as in pressure joints, not being essential. The conditions previously given for the latter, as to plate-rupture and room for the die, hold with regard to the minimum margin and pitch. The maximum pitch in a compression member is fixed by the consideration that the plate in a pitch section is practically a column. In general, also, the maximum pitch must not be so great as to permit the entrance of moisture which would rust and burst the joint. The rivets in the ends of a compression member carry the full load on the member and are spaced with this consideration in view. The specifications of this company as to pitch and margin are:

The pitch of rivets, in the direction of the strain, shall never exceed 6 inches, nor 16 times the thickness of the thinnest outside plate connected, and not more than 40 times that thickness at right angles to the strain.

At the ends of compression members, the pitch shall not exceed 4 diameters of the rivet for a length equal to twice the width of the member.

The distance from the edge of any piece to the centre of a rivet-hole must not be less than 1.5 times the diameter of the rivet nor exceed 8 times the thickness of the plate; and the distance between centres of rivet-holes shall not be less than 3 diameters of the rivet.

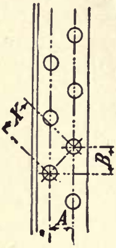
In structural work, the pitch of the rivets may vary between the minimum limit, fixed by the possible cracking of the plate in punching or riveting and the clearance for tools, and the maximum limit

(6″), determined by the union of the parts so that they shall be stressed as a whole and also by the necessity for excluding moisture. Table LII. gives various pitches for double *staggered riveting* and for the staggered spacing of two rows of rivets in the two legs of an angle.

Table LII.

Staggering of Rivets. (American Bridge Co.)

Distance c to c of Staggered Rivets.

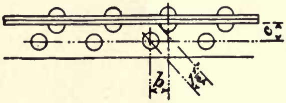


Values of X for Varying Values of A and B.

Values of B.	Values of A.													
	7/8″	1″	1 1/8″	1 1/4″	1 3/8″	1 1/2″	1 5/8″	1 3/4″	1 7/8″	2″	2 1/8″	2 1/4″	2 3/8″	2 1/2″
1 1/8″	1 7/16	1 1/2	1 9/16	1 11/16	1 3/4	1 7/8	2	2 1/16	2 3/16	2 5/16	2 3/8	2 1/2	2 5/8	2 3/4
1 1/4	1 9/16	1 5/8	1 11/16	1 3/4	1 7/8	1 15/16	2 1/16	2 1/8	2 1/4	2 3/8	2 7/16	2 9/16	2 11/16	2 13/16
1 3/8	1 5/8	1 11/16	1 3/4	1 7/8	1 15/16	2	2 1/8	2 3/16	2 5/16	2 7/16	2 1/2	2 5/8	2 3/4	2 7/8
1 1/2	1 3/4	1 13/16	1 7/8	1 15/16	2	2 1/8	2 3/16	2 5/16	2 3/8	2 1/2	2 5/8	2 11/16	2 13/16	2 15/16
1 5/8	1 7/8	1 7/8	2	2 1/16	2 1/8	2 3/16	2 5/16	2 3/8	2 1/2	2 9/16	2 11/16	2 3/4	2 7/8	3
1 3/4	1 15/16	2	2 1/16	2 1/8	2 3/16	2 5/16	2 3/8	2 7/16	2 9/16	2 5/8	2 3/4	2 7/8	2 15/16	3 1/16
1 7/8	2 1/16	2 1/8	2 3/16	2 1/4	2 5/16	2 3/8	2 1/2	2 9/16	2 5/8	2 3/4	2 13/16	2 15/16	3	3 1/8
2	2 3/16	2 1/4	2 5/16	2 3/8	2 7/16	2 1/2	2 9/16	2 5/8	2 3/4	2 13/16	2 15/16	3	3 1/8	3 3/16
2 1/8	2 5/16	2 5/16	2 3/8	2 7/16	2 1/2	2 5/8	2 11/16	2 3/4	2 13/16	2 15/16	3	3 1/16	3 3/16	3 1/4
2 1/4	2 7/16	2 7/16	2 1/2	2 9/16	2 5/8	2 11/16	2 3/4	2 7/8	2 15/16	3	3 1/16	3 3/16	3 1/4	3 3/8
2 3/8	2 1/2	2 9/16	2 5/8	2 11/16	2 3/4	2 13/16	2 7/8	2 15/16	3	3 1/8	3 3/16	3 1/4	3 3/8	3 7/16
2 1/2	2 5/8	2 11/16	2 3/4	2 13/16	2 7/8	2 15/16	3	3 1/16	3 1/8	3 3/16	3 1/4	3 3/8	3 7/16	3 9/16

Note: Values below or to right of upper zigzag lines are large enough for 3/4 rivets.
" " " " " " lower " " " " " ". 7/8 "

Minimum Stagger for Rivets.



c	3/4″ Diam. b	7/8″ Diam. b
1 1/8″	1 1/4″	1 3/8″
1 3/16	1 3/16	1 5/16
1 1/4	1 1/8	1 1/4
1 5/16	1 1/16	1 3/16
1 3/8	15/16	1 1/8
1 7/16	7/8	1
1 1/2	3/4	15/16
1 9/16	5/8	13/16
1 5/8	3/8	11/16
1 11/16	0	1/2
1 3/4	0	5/16

The rivet-spacing for various angles is given by Table LIII. for both longitudinal and transverse pitches. In a "crimped angle," as shown in Fig. 94, the distance, b, should be $1\frac{1}{2}$ in. plus twice

the thickness of chord angles, but never less than 2 in. The clearance required for $\frac{3}{4}$-in. and $\frac{7}{8}$-in. rivets is shown by Fig. 95.

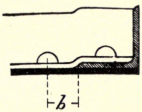


FIG. 94.

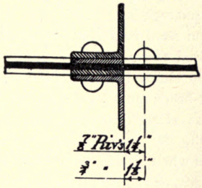
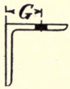
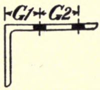


FIG. 95.

TABLE LIII.

RIVET SPACING IN ANGLES. (AMERICAN BRIDGE CO.)





Leg.	G	Max. Rivets.	Leg.	G1	G2	Max. Rivets.
8″	4½″	⅞″	8″	3″	3″	⅞″
7	4	⅞	7	2½	3	⅞
6	3½	⅞	6	2¼	2½	⅞
5	3	⅞	5	2	1¾	⅞
4	2¼	⅞	When 6″ L exceeds ¾″.			
3½	2	⅞	6″	2½″	2¼″	⅞″
3	1¾	⅞				
2¾	1⅝	¾				
2½	1⅜	⅝				
2¼	1¼	⅝				
2	1⅛	½				
1¾	1	½				
1½	⅞	⅜				
1¼	¾	⅜				
1	9/16	¼				

MINIMUM RIVET SPACING.

Size of Rivet.	¼″	⅜″	½″	⅝″	¾″	⅞″	1″
Minimum Distance.	1″	1¼″	1¾″	2″	2¼	2⅝	3″

4. PUNCHING AND RIVETING. — The specifications of this company for steel railroad bridges are:

All riveted work shall be punched accurately with holes $\frac{1}{16}$ in. larger than the size of the rivet, and, when the pieces forming one built member are put together, the holes

must be truly opposite. No drifting to distort the metal will be allowed; if the hole must be enlarged to admit the rivet, it must be reamed.

All holes for field-rivets, excepting those in connections for lateral and sway bracing, shall be accurately drilled to an iron templet or reamed while the connecting parts are temporarily put together.

In medium steel over $\frac{5}{8}$ in. thick, all sheared edges shall be planed and all holes shall be drilled or reamed to a diameter $\frac{1}{8}$ in. larger than the punched holes, so as to remove all the sheared surface of the metal.

The rivet-heads must be of approved hemispherical shape and of a uniform size for the same size rivets throughout the work. They must be full and neatly finished throughout the work and concentric with the rivet-hole.

All rivets when driven must completely fill the holes, the heads be in full contact with the surface or countersunk when so required.

Wherever possible, all rivets shall be machine-driven. Power-riveters shall be direct-acting machines, worked by steam, hydraulic pressure, or compressed air.

5. Stresses in Riveted Members. — The built-up members of framed structures are made of rolled shapes of various forms, plates, angles, etc., joined by rivets which distribute and transfer the stress developed by the load. Rivets should be subjected to shearing and bearing stresses only. The connected parts resist shear, tension, compression, or compound stress, as their location with respect to the load determines.

Working Stresses.—The greatest permissible working stresses for the parts of a *member* vary with the location of the part and the character of its load. For wrought iron, general values in lbs. per sq. in. are: in tension, 7,500; in shear, 6,000. For steel, the tensile stress ranges from 10,000 to 17,000 and the shearing stress from 6,000 to 11,000. For compression members, the permissible stresses are those for tension, modified by the relation between the length and least radius of gyration of the section. One formula of this nature is quoted below.

The shearing resistance in lbs. per sq. in. of cross-section for *rivets* in single shear is: iron, 6,000 to 7,500; steel, 7,500 to 12,000. Corresponding values of the bearing pressure are: iron, 12,000 to 15,000; steel, 15,000 to 24,000 lbs. per sq. in. upon the projected area equal to diameter of rivet-hole $\times$ thickness of plate. For field-riveting, the number of rivets as calculated is increased by 10 to 50 per cent., as a margin for defective work. Table LIV. gives the shearing and bearing values of rivets for resistances of 11,000 lbs. (single shear) and 22,000 lbs. per sq. in. respectively.

TABLE LIV.

SHEARING AND BEARING VALUE OF RIVETS IN POUNDS.

Diameter of Rivet, Inches. Fraction.	Decimal.	Area in Square Inches.	Single Shear at 11,000 Lbs.	Bearing Value for Different Thicknesses of Plate in Inches at 22,000 Pounds per Square Inch. $\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
$\frac{3}{8}$	.375	.1104	1210	2060	2580	3090										
$\frac{1}{2}$	.500	.1963	2160	2750	3440	4130	4820	5500								
$\frac{5}{8}$	.625	.3068	3370	3440	4300	5160	6020	6880	7740	8600						
$\frac{3}{4}$	.750	.4418	4860	4130	5160	6190	7220	8250	9280	10320	11340	12380				
$\frac{7}{8}$	.875	.6013	6610	4810	6020	7220	8430	9630	10840	12040	13240	14440	15640	16840	18050	
1	1.000	.7854	8640	5500	6880	8250	9630	11000	12380	13750	15130	16500	17880	19250	20630	22000

All bearing values above or to right of upper zigzag lines are greater than double shear.
Values below or to left of lower zigzag lines are less than single shear.

The following extracts from specifications refer to the requirements of this company as to working stresses in steel railroad bridges:

All parts of the structure shall be so proportioned that the sum of the maximum loads, together with the impact, shall not cause the tensile strain to exceed: on soft steel, 15,000 lbs. per sq. in.; on medium steel, 17,000 lbs. per sq. in.

* * * For compression members, these permissible strains of 15,000 and 17,000 lbs. per sq. inch shall be reduced in proportion to the ratio of the length to the least radius of gyration of the section by the following formulæ:

$$\text{For soft steel, } p = \frac{15,000}{1 + \frac{l^2}{13,500r^2}}; \tag{128}$$

$$\text{For medium steel, } p = \frac{17,000}{1 + \frac{l^2}{11,000r^2}}, \tag{129}$$

where p = permissible working strain per sq. in. in compression; l = length of piece in inches, centre to centre of connection; r = least radius of gyration of the section in inches.

* * * The shearing strain on rivets, bolts, or pins per sq. in. of section shall not exceed 11,000 lbs. for soft steel and 12,000 lbs. for medium steel; and the pressure upon the bearing surface of the projected semi-intrados (diameter × thickness) of the rivet, bolt, or pin hole shall not exceed 22,000 lbs. per sq. in. for soft steel and 24,000 lbs. for medium steel. In field-riveting, the number of rivets thus found shall be increased 25 per cent., if driven by hand, but 10 per cent. if driven by power.

* * * The shearing strain in web-plates shall not exceed 9,000 lbs. per sq. in. for soft steel and 10,000 lbs. per sq. in. for medium steel; but no web plate shall be less than ⅜in. in thickness.

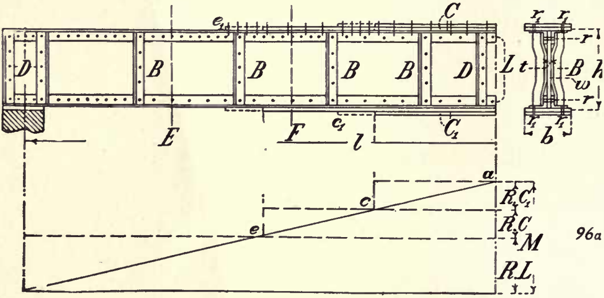


FIG. 96.

6. DISTRIBUTION OF STRESSES.—The character and distribution of the stresses in a member depend upon the function of the latter. The *Plate-Girder* is a fairly comprehensive example of the principles of design in riveted structural work. As shown in Fig. 96,

it consists essentially of a web-plate, *W*, and four angles, *L*, extending from end to end. To these may be added, at top and bottom, a flange-plate, *C*, of partial or full length, and, if required, one or more flange-plates, C_1, traversing the section in which the magnitude of the bending moment makes necessary the additional flange-area. These plates are all of the same width, which is that of the girder. The upper and lower flanges are similar, excepting that one plate of the latter is usually thicker to make up for the loss in net plate-section due to rivet-holes. The parts are joined by rows of rivets, *r*, passing through web and angles and rows, r_1, through angles and flange-plates. As a rule, these are single rows. The method of design is indicated briefly below.*

Length, Width, Depth.—The length, *l*, is the distance between the centres of the end-bearings. Since the upper flange is in compression, it may buckle if weak. Hence, if the unsupported length of the girder exceeds 16 to 20 times its width, *b*, the girder should be given lateral support. The depth, *h*, is the distance between the centres of gravity of the cross-sections of the flanges, each of the latter being made up of two angles and the attached flange-plates. The effective depth may be taken, without material error, as that of the web-plate. To avoid excessive deflection, the least depth should be $\frac{1}{20}$ to $\frac{1}{16}$ of the span. For the economical depth, with regard to weight, Mr. C. W. Bryan, C.E., gives the following formulæ: †

Neglecting moment of resistance of web to bending:

$$x = 1.27\sqrt{\frac{m}{ft}}; \quad (130)$$

Considering moment of resistance of web to bending:

$$x = 1.46\sqrt{\frac{m}{ft}}; \quad (131)$$

in which,

$x =$ depth of girder, ins.;
$m =$ centre-moment, inch-lbs., from dead and live loads;
$f =$ allowable fibre-stress on flanges, lbs. per sq. in.;
$t =$ thickness of web, ins.

* For detailed investigation, see Burr: "Elasticity and Resistance," etc., 1897, p. 578; Johnson, Bryan, Turneaure: "Modern Framed Structures," 1901, p. 292.

† "Modern Framed Structures," 1901, p. 299.

Moments, Vertical Shear, Flange-stress.—The external forces acting on the girder are the loads and the reactions at the supports. These are transmitted directly to the web by vertical angles, D, riveted to the web at the supports and under concentrated loads and by the rivets, r, of the compression flange. The vertical shear produced in the web by the loads acts upon the rivets, r, of both flanges with a leverage equal to the pitch of the rivets, and thus develops bending stress in the flanges. Since the parts are so bound together that the girder bends as a whole, bending stress, in addition to shear, acts in the web. Two methods of design are used: Either to assume that the web is subjected to vertical shear only and proportion the flanges for the full bending stress; or—and correctly—to allow for the resistance to bending of the web and design the flange-area for the remainder of the load.

Let A be the sectional area of the angles and plates (*web not included*) forming one flange at any given point in the girder and S the mean unit working stress over that area. Then, $A \times S$ is the total load or horizontal bending stress on the flange and $A \cdot S \times \frac{h}{2} =$ resisting moment of this stress about the neutral axis of the girder. Assuming A and S as the same for both flanges and *neglecting the bending stress on the web*, the external bending moment at the given point = resisting moment of girder at that point; or,

$$M = 2\left(A.S.\frac{h}{2}\right) = A.S.h \therefore \qquad (132)$$

$$\textit{Flange-stress} = A.S. = \frac{M}{h}; \qquad (133)$$

$$\textit{Flange-area} = A = \frac{M}{h.S}. \qquad (134)$$

The web-section is that of a rectangular beam of depth, h, and breadth equal to the thickness, t. Hence, its resistance to bending is $\frac{S.th^2}{6}$ To allow for the reduced section due to vertical

rows of rivet-holes the 6 is replaced by 8. Hence, *considering the resistance of the web to bending stress:*

$$M = h.S\left(A + \frac{t.h}{8}\right); \tag{135}$$

$$\textit{Flange-stress} = A.S = \frac{M}{h} - \frac{S.t.h}{8}; \tag{136}$$

$$\textit{Flange-area} = A = \frac{M}{S.h} - \frac{t.h}{8}. \tag{137}$$

For steel girders in buildings, the usual unit flange-stress, S, is 15,000 lbs. per sq. in. and the unit shearing-stress, S_s, on the web is 7,000 to 11,000 lbs. per sq. in.

FLANGE-AREA, ANGLES, FLANGE-PLATES.—The required flange-area at any given point in the girder may be found from (134) or (137). The area found thus, serves for the compression flange which is assumed as not weakened by the rivet-holes, since the rivets should about fill the latter. The resistance of plates and angles in the tension flange is that of their net section. The two rivet-rows, r_1, are opposite each other; but r_1 is staggered with regard to r. Hence, the net section of a cover or flange-plate is $(b - 2d) \times$ thickness and the net area of an angle $=$ gross area $- d \times$ thickness. The diameter, d, is that of the rivet plus $\frac{1}{8}$ in., to allow for enlarged hole and effects of punching. The increased area required in the tension flange is added by thickening one of the flange-plates or by calculating for the tension-flange and making the area of both flanges the same. The flange-area may be calculated for different points in the girder or it may be found graphically as shown in Fig. 96*a*, which is the bending moment diagram for a concentrated load at the centre of the girder. The maximum bending moment, M, should be equal to the sum of the *individual* resisting moments $R.L$, $R.C$, $R.C_1$, of the angles, L, and the flange-plates, C and C_1, *i. e.*, on the same scale, $M = R.L + R.C + R.C_1$. Hence, at a, the full section will be required; at c, that of L and C; and at e that of L only. These theoretical lengths of flange-plates are increased somewhat, as will be shown later. In general, not less than 50 per cent. of the maximum flange-area should be in the angles, since the thinner the flange-plates, the

less their leverage through the rivets on the angles, both vertically and with regard to the centre of gravity of the latter.

The sectional areas of various angles are given in Table LV. The centre of gravity of a given flange-area should be as far as possible from the neutral axis of the girder in order to give the maximum resisting moment to bending and the maximum breadth of the area should be as great as the conditions permit in order to strengthen the girder against buckling or lateral yielding. Hence, the angles should have unequal legs with the longer horizontal. The thickness of the angle should be, approximately, that of the web-plate.

Table LV.

Angles: Sectional Areas (sq. ins.).

(American Bridge Co.)

Size.	$\frac{1}{8}''$	$\frac{3}{16}''$	$\frac{1}{4}''$	$\frac{5}{16}''$	$\frac{3}{8}''$	$\frac{7}{16}''$	$\frac{1}{2}''$	$\frac{9}{16}''$	$\frac{5}{8}''$	$\frac{11}{16}''$	$\frac{3}{4}''$	$\frac{13}{16}''$	$\frac{7}{8}''$	$\frac{15}{16}''$	$1''$
$8'' \times 8''$							7.76	8.76	9.76	10.76	11.47	12.47	13.47	14.50	15.53
6×6					4.35	5.09	5.79	6.47	7.18	7.79	8.47	9.12	9.82	10.56	
5×5					3.62	4.21	4.79	5.35	5.91	6.47	7.00	7.53	8.06	8.65	
4×4				2.41	2.88	3.32	3.76	4.21	4.65	5.06	5.47				
$3\frac{1}{2} \times 3\frac{1}{2}$				2.09	2.50	2.88	3.26	3.65	4.03						
3×3			1.44	1.79	2.12	2.44	2.76	3.06	3.38						
$2\frac{3}{4} \times 2\frac{3}{4}$			1.32	1.65	1.94	2.26	2.53								
$2\frac{1}{2} \times 2\frac{1}{2}$		0.91	1.21	1.47	1.74	2.03	2.29								
$2\frac{1}{4} \times 2\frac{1}{4}$		0.79	1.06	1.32	1.59										
2×2		0.74	0.94	1.18	1.41										
$1\frac{3}{4} \times 1\frac{3}{4}$		0.62	0.82	1.03	1.21										
$1\frac{1}{2} \times 1\frac{1}{2}$	0.35	0.53	0.71	0.85	1.03										
$1\frac{1}{4} \times 1\frac{1}{4}$	0.29	0.44	0.59												
1×1	0.24	0.35	0.44												
8×6							6.76	7.59	8.44	9.32	9.94	10.76	11.62	12.50	13.41
$7 \times 3\frac{1}{2}$							5.00	5.59	6.18	6.76	7.29	7.85	8.41	8.97	9.56
6×4					3.59	4.21	4.79	5.32	5.91	6.47	7.00	7.53	8.06	8.65	
$6 \times 3\frac{1}{2}$					3.41	4.00	4.56	5.03	5.59	6.12	6.65	7.21	7.79	8.41	
5×4					3.24	3.76	4.29	4.76	5.26	5.76	6.26				
$5 \times 3\frac{1}{2}$				2.56	3.03	3.53	4.00	4.47	4.94	5.41	5.88				
5×3				2.41	2.85	3.29	3.76	4.18	4.62	5.06	5.50				
$4 \times 3\frac{1}{2}$				2.26	2.68	3.09	3.50	3.91	4.32	4.71	5.12				
4×3				2.09	2.50	2.88	3.26	3.65	4.06						
$3\frac{1}{2} \times 3$				1.94	2.29	2.68	3.03	3.41	3.79						
$3\frac{1}{2} \times 2\frac{1}{2}$			1.44	1.79	2.12	2.44	2.76								
$3 \times 2\frac{1}{2}$			1.32	1.62	1.94	2.26	2.56								
$2\frac{1}{2} \times 2$		0.79	1.06	1.32	1.59	1.82	2.06								
$2 \times 1\frac{1}{2}$		0.62	0.85	1.06	1.26										
$2 \times 1\frac{1}{4}$		0.56	0.76	0.97	1.15										

In general, the aim in design should be to make the girder as deep and the angles as heavy as possible in order to reduce the thickness of the plates and their consequent leverage through

the rivets on the angles. In compression, one plate is better than two of the same aggregate thickness. To increase the resistance to buckling, the plates of the compression flange may be made the full length of the girder. Since the stress is transmitted to the plates by the flange rivets, r_1, the width of the plates outside of those rows is limited by the necessity for an approximately uniform distribution of stress.

Web-plate, Stiffeners. — The thickness of the web-plate, W, must be such as to provide for the maximum vertical shear at the supports and to give sufficient bearing area for the rivets joining the web and angles without making the pitch of the rivets smaller than the minimum allowable. The minimum thickness of web is $\frac{1}{4}$ in. for light work. In railway bridges, no web less than $\frac{3}{8}$ in. thick should be used.

The load on the web not only produces vertical shear and bending stress but also tends to make the plate yield vertically by buckling. The latter is prevented by pairs of vertical angles or stiffeners, B, riveted to the web. If the thickness of the latter is less than $\frac{1}{60}$ of its depth, the stiffeners are placed at intervals not greater than the depth of the girder throughout the length of the latter, with a maximum spacing of 5 ft. Angles for stiffening solely may be made very light. They should bear against both upper and lower flange angles and the web, being bent inward to the latter or left straight and a filling piece interposed.

At the supports and under concentrated loads, as at D, the stiffeners have the further function of transferring the external loads to the web-plate. The rivets passing through them should have a strength sufficient for this. Hence, in addition to the fitting and bearing as above, these transferring stiffeners should be broad enough to give space for the rivets and thick enough to prevent the pressure on the rivets and of the stiffener on the lower flange-angle from exceeding the allowable bearing stress.

Riveting. — The rivets r, and r_1, are in double and single shear respectively and both are under bearing pressure which must be computed for the least bearing surface in either direction of stress. The strength, s, of a rivet is its least resistance under either of the two stresses to which it is subjected. The vertical shear in the web acts through the rivets, r, on the flanges, bending the latter. *Considering only bending stress*, the required number of the rivets, r, depends upon their diameter and the magnitude of the bending

moment. Let M_e be the bending moment on the tension flange at E. The flange-stress $A_e.S$ divided by the strength, s, will give the number of rivets for M_e from E to either support. Again, the increase of moment between E and F is $M_f - M_e$ and the increase of flange-stress is $S(A_f - A_e)$. The latter divided by s gives the number of rivets to be added between E and F, the total number thus far being that on either side of F for the moment M_f. In general, the number, N, of rivets required on each side for a moment, M, is:

$$N = \frac{A.S}{s}. \tag{138}$$

Again, from the relation between the bending moment and the vertical shear, V, we have, for an elementary length, dx:

$$V = \frac{dM}{dx} \therefore V \times dx = dM,$$

i. e., the vertical shear at the left of dx acts with a leverage dx to produce the increment of moment, dM. Let the length of the section be the pitch, p, between two rivets, a at the left and b at the right. Then, if M_a, A_a, M_b, and A_b be the respective moments and flange-areas, we have, *neglecting resistance to bending of web:*

$$V \times p = M_b - M_a = h\,[A_bS - A_aS] = h.s\,;$$

$$p = \frac{h.s}{V}, \tag{139}$$

since the rivet, b, must resist the increment of flange-stress required in the distance, p.

In the upper or compression flange, in addition to the horizontal bending-stress, the rivets, r, are subjected to *vertical stress from the loads transmitted by them to the web.* These loads are: the weight of the girder, the uniform load if any, and any concentrated load which is not provided for fully by transmitting angles, D. Let w be the sum of these loads per inch of length of girder at the section considered. Then $p \times w$ will be the vertical load upon the pitch section and on one rivet. From (139) the horizontal load due to any pitch, p, is $\frac{Vp}{h}$. *The final stress upon the rivet* will be the resultant of these two loads which are normal

to each other and this resultant must not exceed the strength, s, of the rivet. Hence, *neglecting resistance to bending of web:*

$$s^2 = \frac{V^2p^2}{h^2} + p^2w^2;$$

$$p = s.h\sqrt{\frac{1}{V^2 + h^2w^2}},^* \qquad (140)$$

p and h being expressed in inches and s, V, and w in lbs.

From (139) and (140) the pitches in both flanges of rivets, r, may be obtained. If the bending resistance of the web be considered, equations (139) and (140) must be modified in accordance with the terms of (136). The shear is greatest and the pitch least at the supports, both varying in some degree at every section. In practice, the minimum pitch required is generally preserved until the maximum (6 in.) can be used. The number of rivets in the tension flange will be less than that in the upper angle, but, for constructive reasons, rivets inserted below should be in line vertically with those above. When the moment of resistance of the web to bending is considered, the pitch formulæ should be changed to include this factor.

The pitch of the rivets, r_1, joining the flange-plates to the angles is 6 in. excepting at the ends of the plates. The latter are extended beyond the theoretical limits sufficiently to provide space for enough rivets in the two rows to carry the load on the plate in each case, the pitch of these rivets being 4 diameters. Thus, theoretically, the plate, C, ends at e. The load on this plate is approximately the product of its net cross-section and the working stress. Dividing this load by the strength of one rivet, we have the *total* number of rivets in *both* rows to be driven between e and the end, e_1, of the plate.

The following extracts from the specifications of this company for steel railroad bridges cover the points discussed above:

"Girders shall be proportioned on the assumption that $\frac{1}{8}$ of the gross area of the web is available as flange-area. The compressed flange shall have the same sectional area as the tension flange; but the unsupported length of flange shall not exceed 16 times its width.

"In calculating shearing strains and bearing strains on web rivets of plate-girders, the whole of the shear acting on the side next the abutment is to be considered as being transferred into the flange angles in a distance equal to the depth of the girder.

"The web shall have stiffeners riveted on both sides, with a close bearing against upper and lower flange angles, at the ends and inner edges of bearing plates and at all

* "Modern Framed Structures," 1901, p. 306.

points of local and concentrated loads; and also when the thickness of the web is less than $\frac{1}{60}$ of the unsupported distance between flange-angles, at points throughout the length of the girder generally not farther apart than the depth of the full web-plate, with a maximum limit of 5 ft.

"* * * All joints in riveted work, whether in tension or compression members must be fully spliced.

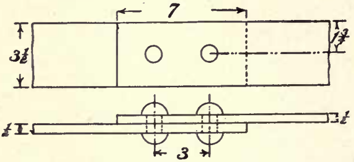
"* * * Web-plates of girders must be spliced at all joints by a plate on each side of the web, not less than $\frac{3}{8}$ in. thick, capable of transmitting the full strain through splice rivets.

"The flange-plates of all girders must be limited in width so as not to extend beyond the outer lines of rivets connecting them with the angles more than 5 in. or more than 8 times the thickness of the first plate. Where two or more plates are used on the flanges, they shall either be of equal thickness or shall decrease in thickness outward from the angles."

TABLE LVI.

RIVETED VS. BOLTED JOINTS.

(LAP JOINTS.)



Joints *A*. Joints *B*.	Double Riveted. Double Bolted.	Treble Riveted. Treble Bolted.	Quadruple Riveted. Quadruple Bolted.
Rivets or bolts, Diam.	$\frac{3}{4}''$	$\frac{3}{4}''$	$\frac{3}{4}''$
" " " No.	2	3	4
" " " Pitch.	3″	3″	3″
Plates, width.	$3\frac{1}{2}''$	$5\frac{1}{2}''$	7″
" thickness.	$\frac{1}{2}''$	$\frac{1}{2}''$	$\frac{1}{2}''$
" lap.	7″	10″	13″
" tensile stress per sq. in. at failure for:			
Iron rivets.	26,120 lbs.	24,310 lbs.	26,450 lbs.
Steel rivets.	26,990 "	29,590 "	28,820 "
Iron bolts.	18,690 "	18,090 "	20,470 "
Steel bolts.	21,110 "	21,460 "	22,060 "

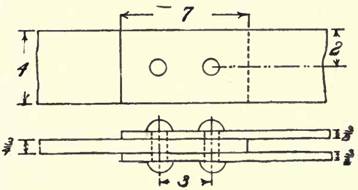
Failure, in all cases, by shearing rivets or bolts.

7. BOLTS. — For facility in erection or in cases where a rivet would be in tension, bolts are used as permanent fastenings in some parts of structural work. For full strength they require to be finished and fitted accurately in drilled holes. Tables LVI. and LVII. give comparative tests — made for the Berlin Iron Bridge Co. at the Watertown Arsenal in 1896 — of riveted and bolted single- and double-shear joints with punched holes. The plates

were of steel; diameter of punch, $\frac{13}{16}$ in., of die, $\frac{7}{8}$ in.; chain-riveting, the "pitch" in the tables being the distance between the rows. The test-piece consisted of a section of the joint containing one rivet in each row. The joints were similar throughout, excepting that the rivets in *A* and *C* were replaced in *B* and *D* by through bolts and nuts.

TABLE LVII.

RIVETED VS. BOLTED JOINTS.
(ONE WEB AND TWO COVER-PLATES.)



Joints *C*. Joints *D*.	Double Riveted. Double Bolted.	Treble Riveted. Treble Bolted.	Quadruple Riveted. Quadruple Bolted.
Rivets or bolts, Diam.	$\frac{3}{4}''$	$\frac{3}{4}''$	$\frac{3}{4}''$
" " " No.	2	3	4
" " " Pitch.	3″	3″	3″
Plates, width.	4″	6″	8″
" thickness.	$\frac{3}{4}$ and $\frac{3}{8}''$	$\frac{3}{4}$ and $\frac{3}{8}''$	$\frac{3}{4}$ and $\frac{3}{8}''$
" lap.	7″	10″	13″
" tensile stress. (web) per sq. in. at failure for:			
Iron rivets.	29,670 lbs.	28,520 lbs.	28,470 lbs.
Steel "	29,540 "	31,040 "	28,970 "
Iron bolts.	18,000 "	17,720 "	18,990 "
Steel "	20,050 "	25,010 "	23,440 "
Manner of failure for:			
Iron rivets.	Sheared rivets.	Sheared rivets.	Sheared rivets.
Steel "	Fractured web plate.	" "	Fractured covers through rivet holes.
Iron bolts.	Sheared bolts, both planes.	" bolts.	Sheared bolts.
Steel "	Sheared bolts, one plane.	" "	" "

53. Riveted Joints, Hull Plating.

The hull of a ship is essentially a girder constructed to bear a given maximum load with various modes of support. Hence the principles which govern the design of structural work in

general apply to the proportion and connection of the members of hull-framing. In the joints of the outside plating and those of the double bottom, bulkheads, etc., there is the further requirement of tightness against water-pressure. Since the latter is, in any event, but moderate, these joints hold an intermediate position, with respect to tightness, between those of general structural work and the seams of steam boilers.

In outside plating, the longitudinal seams are lapped except where flush work is required. The transverse seams are butt joints with single straps. Rivet-points are countersunk and chipped and all seams are calked. The riveting is done either by hand-work, or by portable hydraulic or pneumatic machines carried on a gantry which spans the ship, or by the pneumatic riveting hammer which, with its frame and pneumatic "holder-on," forms a readily portable combination operated by two men. The latter method for hull-riveting is meeting wide adoption in the United States. With regard to its cost and results as compared with hand-work, Edwin S. Cramp, Esq., Vice President of the William Cramp and Sons Ship and Engine Building Co., says:

"We have found that the use of these tools results in an increase of operating expenses and a decrease in labor-charges, with a net saving of about ten per cent. over the cost of hand-riveting. The quality of the work done and the speed with which it is done are increased to a great extent."

Rivets are usually of steel. Up to $\frac{3}{8}$ in. diameter they should be riveted cold, since the rivet-blank of small size not only cools very quickly but, proportionately, wastes much more rapidly by oxidation and scaling. For cold-riveting, the steel should be soft and ductile as the harder metal of higher tensile strength becomes brittle and untrustworthy when thus worked. The countersunk points used in shell plating, while more costly and giving a reduced net section of plate, have two great advantages: Their use removes much of the metal injured in punching the holes and they add no weight to that of the full plate, since they are chipped flush. Weight-saving without reduction of strength is a matter of importance in ships, especially in men-of-war, since useless weight is but so much unprofitable load to be transported during the life of the ship. Rivets form a very considerable proportion of the total weight of the hull. Naval Constructor J. H. Linnard, U. S. Navy, gives,* for the U. S. Armored Cruiser *Brooklyn* (9,270 tons), the

* *Trans. Soc. Naval Architects and Marine Engineers*, Vol. IV.

total weight of rivets driven as upward of 330,000 lbs., of which weight from $\frac{1}{4}$ to $\frac{1}{3}$ is in the rivet-heads and points. The rivet blank will have a better fit in punched holes which are not countersunk, if coned under the head as shown in Fig. 97.

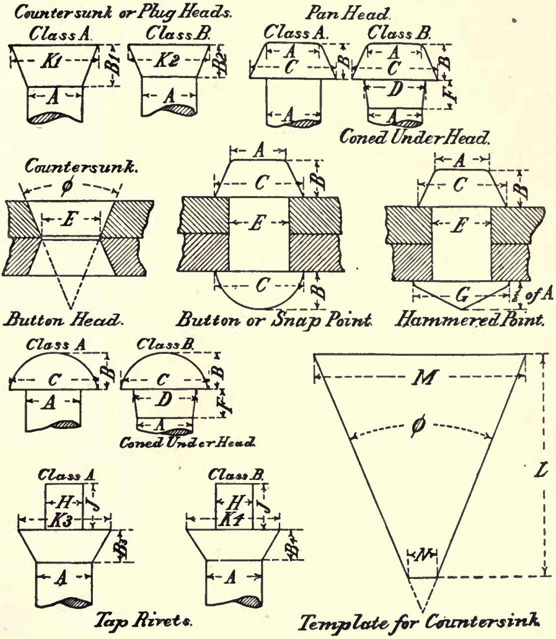


FIG. 97.

1. U. S. NAVAL PRACTICE.—The following extracts, relating to hull-riveting *in general*, are taken from the Specifications (1899) of the Bureau of Construction and Repair, U. S. Navy.

PLATE AND RIVET METALS:

SHIP PLATES AND SHAPES.

26. *Kind of Material.*—Plates and shapes shall be of steel or nickel steel, made by the open hearth process, and must not show more than *six one-hundredths* (.06) of one per cent. of phosphorus, nor more than *four one-hundredths* (.04) of one per cent.

of sulphur for acid steel; and not more than *four one-hundredths* (.04) of one per cent. of phosphorus, nor more than *four one-hundredths* (.04) of one per cent. of sulphur for basic steel. The material shall be of the best composition in all other respects.

At the option of the manufacturer the material may be annealed.

The material will be classified in three standard grades according to its characteristics and the purposes for which intended. These grades will be known as (1) soft or flange steel, (2) medium steel and (3) hard steel.

The following are the minimum requirements of each grade of steel:

Grade.	Tensile Strength.	Elongation.	Cold Bend.	Quench Bend.
Soft or flange steel.	50,000 lbs.	30 per cent.	180° flat.	180° flat.
Medium steel.	60,000 lbs.	25 per cent.	For plates below $\frac{3}{4}$ inch in thickness: 180° flat for longitudinal; 180° to diameter of 1 thickness for transverse. For plates above $\frac{3}{4}$ inch in thickness, the bends will be 180° to a diameter of 1 thickness for longitudinal, and 2 thicknesses for transverse specimens.	For plates below $\frac{3}{4}$ inch in thickness; 180° to diameter of $1\frac{1}{2}$ thicknesses for longitudinal; 180° to diameter of $2\frac{1}{2}$ thicknesses for transverse. For plates above $\frac{3}{4}$ inch in thickness, the bends will be 180° to a diameter of $1\frac{1}{2}$ thicknesses for longitudinal, and $2\frac{1}{2}$ thicknesses for transverse specimens.
Hard steel.	75,000 lbs.	18 per cent.	180° to a diameter of $1\frac{1}{2}$ thicknesses for longitudinal; 180° to a diameter of 3 thicknesses for transverse.	No quench bend.

Protective Deck Plating.

38. The lower courses of plating for the protective deck will be of steel of the qualities of ship plate, and shall be inspected accordingly.

39. The upper course of plating of protective deck shall be of nickel steel, containing about *three and a quarter* ($3\frac{1}{4}$) per cent. of nickel, not more than *six one-hundredths* (.06) of 1 per cent. of phosphorus, nor more than *four one-hundredths* (.04) of 1 per cent. of sulphur, and be of the best composition in all other respects. All these plates shall be oil- or water-tempered and annealed.

All rivets are of steel whose characteristics are:

Hull Rivets.

43. *Kind of Material.* — Steel for hull rivets shall be made by the open-hearth process, and not show more than *five one-hundredths* (.05) of one per cent. of phosphorus, nor more than *four one-hundredths* (.04) of one per cent. of sulphur, and be of the best composition in other respects.

44. A whole heat or part thereof may be rolled into rivet rods, and from each size rolled six (6) tensile tests shall be taken at random, each from a different bar as finished at the rolls. When lots smaller than five tons are rolled, one test piece shall be taken from each size for every ton or fraction thereof.

45. Rods from which rivets are to be made of a diameter of one half ($\frac{1}{2}$) inch or less shall be tested in the diameter of the finished rivet. These rods shall show a tensile strength of not less than 58,000 pounds per square inch and an elongation of not less than 28 per cent.

46. Rods from which rivets are to be made of a diameter above one half ($\frac{1}{2}$) inch shall be tested with specimens of the same diameter as the finished rivet, when practicable, and shall show the same tensile strength as the smaller rivets and an elongation of not less than 29 per cent. Specimens from these rods shall be of the maximum cross section within the capacity of the testing machine.

47. From each lot of rivets kegged and ready for shipment there shall be taken at random six (6) rivets, to be tested as follows:

(*a*) Three rivets to be flattened out cold under the hammer to a thickness of one half ($\frac{1}{2}$) the diameter of the part flattened, without showing cracks or flaws. Rivets of over an inch in diameter shall be flattened to three fourths ($\frac{3}{4}$) of the original diameter.

(*b*) Three rivets to be flattened out hot under the hammer to a thickness of at least one third ($\frac{1}{3}$) of the original diameter of the part flattened, the heat to be the ordinary driving heat.

(*c*) From each heat of rivet rods as finished at the rolls four cold-bending tests shall be taken, which shall be bent over flat on themselves without showing any cracks or flaws on the outer round.

48. *Inspection for Surface and Other Defects.* — Rivets must be true to form, concentric, and free from scale, fins, seams, and all other injurious or unsightly defects. Tap rivets will be milled under the head, if necessary to obtain accuracy.

49. The style of rivet used will be determined by the Superintending Naval Constructor. As a general rule, countersunk *heads* will be used *only* where required by mechanical or other special reasons. Wherever practicable, the pan-head rivet will be used, with countersunk points where flush work is required, button or snap points for finished appearance or where rivets are closed by power and hammered or mashed points elsewhere. Where points are made with a snap-tool, or where riveting by power is employed, care will be taken that the points are properly centered. In these cases, and also in the case of hammered points, the aim must be to have the point of adequate strength, following as nearly as possible the dimensions of points given in Table LVIII. Care must be taken that snap points are not reduced from the standard sizes by grinding down tools that have been chipped or burred. All pan-head rivets not less than ½ inch diameter for punched holes should be coned under head, as shown in Fig. 97, the rule for punching from the faying surface of plate being carefully observed. If, however, the practice of punching the holes small and reaming to size by power be employed, the coning under head may be omitted.

Proportions of Seams. — General proportions of plates, laps, straps, rivets, and spacing are given in Table LVII, page 240.

Proportions of Rivets. — The approved types of *head and points* for torpedo-boats and ship work are given in Fig. 97, page 237, and Table LVIII, page 241.

TABLE LVII.

COMBINATION TABLE FOR SHIP WORK.

Gauge of Plates.	Corresponding Thicknesses.	Diameter of Rivet.	Diameter of Hole.	Breadth of Laps.						Width of Strips and Single Straps.		
				Single Riveting.	Double Chain Riveting.	Double Zigzag Riveting.	Treble Chain Riveting.	Double-Riveted Butt Laps.	Treble-Riveted Butt Laps.	Single Riveting.	Double Riveting.	Treble Riveting.
Pounds per sq. ft.	*Thirty-seconds of an inch.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
Up to 3	Up to 2	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{13}{16}$	$1\frac{7}{16}$	$1\frac{1}{4}$	$2\frac{1}{16}$	$1\frac{9}{16}$	$2\frac{5}{16}$	$1\frac{5}{8}$	$2\frac{7}{8}$	$4\frac{1}{8}$
3 to 6	2 to 5	$\frac{3}{8}$	$\frac{7}{16}$	$1\frac{3}{16}$	$2\frac{1}{8}$	$1\frac{7}{8}$	$3\frac{1}{16}$	$2\frac{5}{16}$	$3\frac{7}{16}$	$2\frac{7}{16}$	$4\frac{5}{16}$	$6\frac{3}{16}$
6 to 8	5 to 7	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{5}{8}$	$2\frac{7}{8}$	$2\frac{1}{2}$	$4\frac{1}{8}$	$3\frac{1}{8}$	$4\frac{5}{8}$	$3\frac{1}{4}$	$5\frac{3}{4}$	$8\frac{1}{4}$
8 to 13	7 to 11	$\frac{5}{8}$	$\frac{11}{16}$	$2\frac{3}{16}$	$3\frac{5}{8}$	$3\frac{1}{8}$	$5\frac{3}{16}$	$3\frac{15}{16}$	$5\frac{13}{16}$	$4\frac{1}{16}$	$7\frac{3}{16}$	$10\frac{5}{16}$
13 to 20	11 to 15	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{7}{16}$	$4\frac{5}{16}$	$3\frac{3}{4}$	$6\frac{3}{16}$	$4\frac{3}{4}$	$6\frac{15}{16}$	$4\frac{7}{8}$	$8\frac{5}{8}$	$12\frac{3}{8}$
20 to 30	15 to 24	$\frac{7}{8}$	$\frac{15}{16}$		5	$4\frac{3}{8}$	$7\frac{3}{16}$	$5\frac{1}{2}$	$8\frac{1}{16}$	$5\frac{3}{4}$	$10\frac{1}{16}$	$14\frac{7}{16}$
30 to 40	24 to 32	1	$1\frac{1}{16}$		$5\frac{3}{4}$	5	$8\frac{1}{4}$	$6\frac{1}{4}$	$9\frac{1}{4}$	$6\frac{1}{2}$	$11\frac{1}{2}$	$16\frac{1}{2}$
40 to 51	32 to 41	$1\frac{1}{8}$	$1\frac{7}{32}$				$9\frac{1}{4}$	7	$10\frac{3}{8}$	$7\frac{5}{16}$	$12\frac{15}{16}$	$18\frac{9}{16}$
51 and over.	41 and over.	$1\frac{1}{4}$	$1\frac{11}{32}$				$10\frac{5}{16}$	$7\frac{13}{16}$	$11\frac{9}{16}$	$8\frac{1}{8}$	$14\frac{3}{8}$	$20\frac{5}{8}$

TABLE LVIII.

PROPORTIONS OF RIVETS.

	Size of Rivet.	E	A	B	C	D	F	G	B_1	K_1	B_2	K_2	B_3	K_3	B_4	K_4	H	J	ϕ	L	M	N
Torpedo Boats.	$\frac{1}{4}$ inch.	$\frac{9}{32}''$	$\frac{1}{4}''$	$\frac{3}{16}$	$\frac{7}{16}$			$\frac{1}{2}$	$\frac{3}{32}$										60°	$2\frac{1}{2}''$	$3\frac{1}{64}$	$\frac{1}{8}$
	$\frac{5}{16}$ "	$\frac{11}{32}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{1}{2}$			$\frac{5}{8}$	$\frac{3}{16}$										60	"	"	"
	$\frac{3}{8}$ "	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{9}{16}$			$\frac{3}{4}$	$\frac{7}{32}$										60	"	"	"
	$\frac{7}{16}$ "	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{11}{32}$	$\frac{11}{16}$			$\frac{7}{8}$	$\frac{1}{4}$										60	"	"	"
	$\frac{1}{2}$ "	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{9}{16}$	$\frac{1}{4}$	1	$\frac{5}{16}$										60	"	"	"
	$\frac{5}{8}$ "	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	1	$\frac{11}{16}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{3}{8}$										60	"	"	"
Ship Work.	$\frac{1}{4}$ inch.	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{7}{16}$			$\frac{1}{2}$	$\frac{3}{32}$	$\frac{11}{32}$									53	$2\frac{1}{2}$	$2\frac{5}{8}$	$\frac{1}{8}$
	$\frac{3}{8}$ "	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{8}$			$\frac{3}{4}$	$\frac{3}{16}$	$\frac{9}{16}$									53	"	"	"
	$\frac{1}{2}$ "	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{15}{16}$	$\frac{1}{4}$	$\frac{3}{4}$									53	"	"	"
	$\frac{5}{8}$ "	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	1	$\frac{11}{16}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{3}{8}$	1	$\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{16}$	$\frac{15}{16}$			$\frac{7}{16}$ Sq	$\frac{5}{8}$	53	"	"	"
	$\frac{3}{4}$ "	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{16}$	$\frac{13}{16}$	$\frac{3}{8}$	$1\frac{5}{16}$	$\frac{9}{16}$	$1\frac{3}{16}$	$\frac{7}{16}$	$1\frac{3}{32}$	$\frac{7}{16}$	$1\frac{1}{4}$			$\frac{1}{2}$ "	$\frac{5}{8}$	45	3	$2\frac{7}{8}$	$\frac{3}{8}$
	$\frac{7}{8}$ "	$\frac{15}{16}$	$\frac{7}{8}$	$\frac{9}{16}$	$1\frac{5}{16}$	$\frac{15}{16}$	$\frac{7}{16}$	$1\frac{1}{2}$	$\frac{3}{4}$	$1\frac{7}{16}$	$\frac{9}{16}$	$1\frac{5}{16}$	$\frac{9}{16}$	$1\frac{5}{16}$	$\frac{7}{16}$	$1\frac{7}{32}$	$\frac{9}{16}$ "	$\frac{23}{32}$	45	"	"	"
	1 "	$1\frac{1}{16}$	1	$\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{1}{16}$	$\frac{1}{2}$	$1\frac{3}{4}$	1	$1\frac{5}{8}$	$\frac{5}{8}$	$1\frac{13}{32}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$1\frac{13}{32}$	$\frac{5}{8}$ "	$\frac{23}{32}$	37	"	$2\frac{1}{2}$	$\frac{1}{2}$
	$1\frac{1}{8}$ "	$1\frac{7}{32}$	$1\frac{1}{8}$	$\frac{11}{16}$	$1\frac{5}{8}$	$1\frac{3}{16}$	$\frac{9}{16}$	$1\frac{15}{16}$			$\frac{3}{4}$	$1\frac{5}{8}$	$\frac{3}{4}$	$1\frac{5}{8}$			$\frac{5}{8}$ "	$\frac{23}{32}$	37	"	"	"
	$1\frac{1}{4}$ "	$1\frac{11}{32}$	$1\frac{1}{4}$	$\frac{3}{4}$	$1\frac{13}{16}$	$1\frac{5}{16}$	$\frac{5}{8}$	$2\frac{1}{8}$			$\frac{7}{8}$	$1\frac{13}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$			$\frac{11}{16}$ "	$\frac{23}{32}$	37	"	"	"
																			32	"	$2\frac{3}{16}$	"
		Diameter of Holes.	All Rivets.	Pan and Button Heads.	Snap Points.	Cone Under Head.		Hammered Points.	Countersunk Heads.	Class A.	Countersunk Heads.	Class B.	Tap Rivets.	Class A.	Tap Rivets.	Class B.	Tap Rivets.		For special rivets larger than $1\frac{1}{4}''$, the angle of countersink should be 32°.			

The *diameters* of rivets and rivet holes for torpedo-boat and ship work are given in :

TABLE LIX.

DIAMETER OF RIVETS.

Weight of Plates.	Diameter of Corresponding Rivets.	Diameter of Corresponding Rivet Holes.
FOR TORPEDO-BOAT WORK.	*Inches.*	*Inches.*
Up to 3 pounds, exclusive.	$\frac{1}{4}$	$\frac{9}{32}$
3 pounds to 6 pounds, exclusive.	$\frac{5}{16}$	$\frac{11}{32}$
6 pounds to $7\frac{1}{2}$ pounds, exclusive.	$\frac{3}{8}$	$\frac{7}{16}$
$7\frac{1}{2}$ pounds to 9 pounds, exclusive.	$\frac{7}{16}$	$\frac{1}{2}$
9 pounds to 11 pounds, exclusive.	$\frac{1}{2}$	$\frac{9}{16}$
11 pounds to 13 pounds, exclusive.	$\frac{5}{8}$	$\frac{11}{16}$
FOR SHIP WORK.		
Up to 3 pounds, exclusive.	$\frac{1}{4}$	$\frac{9}{32}$
3 pounds to 6 pounds, exclusive.	$\frac{3}{8}$	$\frac{7}{16}$
6 pounds, inclusive, to 8 pounds, exclusive.	$\frac{1}{2}$	$\frac{9}{16}$
8 pounds, inclusive, to 13 pounds, exclusive.	$\frac{5}{8}$	$\frac{11}{16}$
13 pounds, inclusive, to 20 pounds, exclusive.	$\frac{3}{4}$	$\frac{13}{16}$
20 pounds, inclusive, to 30 pounds, exclusive.	$\frac{7}{8}$	$\frac{15}{16}$
30 pounds, inclusive, to 40 pounds, exclusive.	1	$1\frac{1}{16}$
40 pounds, inclusive, to 51 pounds, exclusive.	$1\frac{1}{8}$	$1\frac{7}{32}$
51 pounds, and above.	$1\frac{1}{4}$	$1\frac{11}{32}$

NOTE.—The sizes of flanges of angles to which plates are connected in torpedo-boat work may sometimes be such as to require a reduction in the size of the rivet to secure satisfactory workmanship.

For connections between plates of different thicknesses and for tap-rivets, the requirements as to diameter are :

11. In cases where rivets connect plates of different thicknesses, the size of rivet indicated for the greater thickness, with corresponding spacing, will be used where strength is required, and that indicated for the lesser thickness where water-tightness is a special consideration, always provided that the greater thickness is not more than double the lesser.

Where tap rivets must be used they should be $\frac{1}{8}$ inch larger than the corresponding ordinary rivets for the same thickness, excepting taps into heavy forgings or castings, such as stem and stern posts, which should be $\frac{1}{4}$ inch larger. Where strength is required tap rivets must not penetrate *less* than 1 diameter, and should penetrate $1\frac{1}{2}$ diameters when the thickness of metal will allow it.

The following extract and Table LX. refer to the length of rivet necessary to form the point :

19. Special care will be taken in riveting that rivets used are of sufficient length to insure a proper point, the aim being to have the rivet a trifle long, if anything. Such cutting as is necessary should be done while the rivet is still a dull red, and the point finished after further cooling. Allowances for length *over* the thicknesses connected are given in Table LX., *below*, the allowance being for two thicknesses only. An additional allowance of $\frac{1}{16}$ should be added to that given for each additional thickness connected, unless the additional thickness is less than $\frac{3}{16}$ inch, when $\frac{1}{32}$ inch additional allowance should be sufficient. The allowances given in the table are based upon the

employment of hand riveting, and are not sufficient in the case of power riveting, for which an added allowance — about $\frac{1}{8}$ inch — should be made in each case. This table must be used judiciously and not absolutely depended upon.

Table LX.

Allowance in Length of Rivets for Points.

Diameter of rivet.	In. $\frac{1}{4}$	In. $\frac{3}{8}$	In. $\frac{1}{2}$	In. $\frac{5}{8}$	In. $\frac{3}{4}$	In. $\frac{7}{8}$	In. 1	Ins. $1\frac{1}{8}$	Ins. $1\frac{1}{4}$
	Allowance for point, over 2 thicknesses connected.								
Type of point:	In.	In.	In.	In.	In.	In.	In.	In.	Ins.
Countersunk.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{16}$
Hammered.	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	1	$1\frac{1}{16}$
Button.			$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$		

The allowances given above apply only to rivets which fit the holes neatly, as hereinbefore described.

Laps and Straps. — The required thickness of single and double butt straps is given in the following extracts and the breadth of laps and straps in Table LXI.

Table LXI.

Breadth of Laps and Straps.

	Diameters.
Breadth of laps for single riveting.	$3\frac{1}{4}$
Breadth of laps for double chain riveting.	$5\frac{3}{4}$
Breadth of laps for double zigzag riveting.	5
Breadth of double-riveted butt laps.	$6\frac{1}{4}$
Breadth of laps for treble riveting.	$8\frac{1}{4}$
Breadth of treble-riveted butt laps in outside plating.	$9\frac{1}{4}$
Breadth of edge strip for single riveting.	$6\frac{1}{2}$
Breadth of edge strip for double riveting.	$11\frac{1}{2}$
Breadth of butt strap for double riveting.	$11\frac{1}{2}$
Breadth of butt strap for treble riveting.	$16\frac{1}{2}$
Breadth of double butt strap, double-riveted.	$12\frac{1}{2}$
Breadth of double butt strap, treble-riveted.	$18\frac{1}{2}$

Single Straps.

16. Single butt straps and edge strips, when single- or double-riveted, should be of the same thickness as the plates connected, and where the plates connected are of different thickness they should be of the same thickness as the lighter plate. Single butt straps, when treble-riveted, should be $1\frac{1}{4}$ times the thickness of plates which they connect.

Double Butt Straps.

17. Double butt straps should not be used for water-tight work, owing to difficulty in calking. They may be used to advantage, however, in connections requiring great strength, but not water-tightness. The thickness of each strap should be one half the

thickness of plates connected for double-riveted straps and five eighths the thickness for treble-riveted straps. * * * For double butt straps the size of rivet to be used should be as follows:

For plates from 15 to 20 pounds, exclusive, $\frac{5}{8}$-inch rivets.
For plates from 20 to 25 pounds, inclusive, $\frac{3}{4}$-inch rivets.
For plates above 25 pounds, as per Table LIX.

Since double butt straps are, as a rule, used in hull work in joints requiring strength but not tightness, the spacing of the rivets in such joints is that required for maximum efficiency, the assumptions being: Tensile strength of plate, 63,000 lbs. per sq. in.; shearing strength of rivets, 50,000 lbs. per sq. in.; for single straps, plate to be through-countersunk.

Table LXII.

Spacing of Rivets.

Connection.	Number of Diameters from Center to Center.
Single-riveted laps and straps.	$3\frac{1}{2}$
Double-riveted laps and straps.	4
Treble-riveted laps.	$4\frac{1}{2}$
Treble-riveted straps, with alternate rivets in third row omitted.	4
Longitudinal seams of plating required to be water-tight (excepting single-riveted laps and straps).	$4\frac{1}{2}$
Connections of transverse frames not water-tight to outside plating.	8
Connections of deck plating to beams; of nonwater-tight longitudinals to outside plating; of the angles and stiffeners to bulkheads when entirely above the water line, and in general where special strength is not required.	8
Connections of floor plates, brackets, lightened intercostals, etc., to clips and angles; of the vertical keel angles to the flat and vertical keel plates and to the flat keelson plates beyond the limits of double bottom, provided water-tightness is not required.	7
Connections of angles and other stiffeners to bulkheads at or below the water line; of boiler and engine bearings and foundations generally.	6
Connections of inner bottom plating to all frames and longitudinals.	5
Connections of angles of water-tight frames and longitudinals to all plating, and in general where water-tightness is required between shapes and plates.	5
Angles and other stiffeners to bulkheads forming supports to turrets, barbettes, connections of armor shelf angles to plating, etc.	5
Connections between staple angles of water-tight floors and the floor plates.	$4\frac{1}{2}$
In special cases of intercostals, beam ends, etc., where strength is required in connections of limited extent, and in all other exceptional cases, spacing to be as required by circumstances, except that the rivets in the same line should never be spaced less than	3

Rivet-spacing. — General proportions for the spacing of rivets are given in Table LXII. Where this spacing cannot be followed exactly, it may be slightly closer for heavy plates and slightly

wider for light plates. The division between "light" and "heavy" plates lies, with single-riveting, at $7\frac{1}{2}$-lb. plates; with double-riveting at 15-lb. plates; and, with treble-riveting at 25-lb. plates.

15. When strength is required in laps and butted connections of plating, with the spacing indicated, single-riveting is suitable only for plating under $12\frac{1}{2}$ pounds and double-riveting for plating under 25 pounds. * * * For maximum strength in connections of plating above 30 pounds it will generally be found that quadruple-riveting is required.

DISTANCE BETWEEN ROWS.

18. Centers of rivets should be placed not less than $1\frac{5}{8}$ times the diameter from the edges of plates connected. In double- and treble-riveting, for laps and single straps, the distance from centre to centre of rows should not be less than $2\frac{1}{2}$ diameters; in butt laps and double butt straps the distance between centres of rows should be not less than 3 diameters (butt laps should be at least double-riveted). For zigzag riveting the distance between centres of rows should not be less than $1\frac{3}{4}$ diameters for rivets spaced 4 diameters apart in rows.

Punching, Drilling, Riveting.—The size of the rivet-hole for a given diameter of rivet has been given previously.

4. All rivet holes through material 1 inch or more in thickness should be drilled, or, if punched, should afterwards be reamed to finished size. The increase in diameter of hole due to reaming should be equal to at least one eighth the thickness of the material. In punching, where possible, holes will be punched *from* the side which will form the faying surface.

5. Great care must be taken in punching to prevent holes from coming unfair. Any unfair hole must be reamed out before riveting, and a rivet suitable to the increased size of hole inserted.

In countersunk holes, where the depth, *B*, given in Fig. 97, would extend through the plate, the countersink should be carried to within $\frac{1}{32}$ in. of the bottom. Power riveting is preferred for torpedo-boat work. Rivets, in general, less than $\frac{3}{8}$-in. diameter should be driven cold; in torpedo-boat work, $\frac{7}{16}$-in. rivets also may be thus driven.

2. AMERICAN BUREAU OF SHIPPING.—In the rules of this bureau for the building and classing of vessels, the character assigned to the latter is expressed by numerals ranging from *A* 1 to *A* 3, the former being the highest grade and corresponding with the grades *A* 1 of Lloyd's Register and 3/3 — 1.1 of the *Bureau Veritas*. Vessels classified under the latter grades are regarded as fitted for the carriage of all kinds of cargo on all voyages. New vessels built in conformity with, or equal to, the rules of the American Bureau are graded thus: 1st Class, *A* 1 for 17 years; 2d Class, *A* 1 for 13 years; 3d Class, *A* 1 for 10 years. If built under inspection, these terms are increased by three years for the

1st and 2d classes and by two years for all others. The following extracts are taken from the rules (1901) of this Bureau:

Outside (skin) Plating: of all (steel) steam vessels whose length does not exceed 11 times their depth to be, for half the vessel's length amidships, and at ends, the thickness specified in Table LXIII. * * * Skin plates (in general) must not be less than 6 frame spaces in length. * * * Butts in adjoining strakes must be shifted clear of each other not less than two frame-spaces. Butts in alternate strakes must have a clear shift of not less than one frame-space. * * * The butts of all skin-plates must be planed and close fitted and the butt-straps be drawn up iron to iron. * * * The edges of all skin plates to be sheared from their faying surfaces and those of outside strakes to be planed or chipped fair. All butts and seams to be efficiently calked. * * * The skin plating can be worked in out-and-in strakes or flush. If worked in out-and-in strakes, the insides strakes must be fitted to frames, iron to iron, and solid liners must be fitted between the frames and the outside strakes. * * * If the skin plating is worked flush, * * * continuous edge-strips, with their butts shifted well clear of the butts of the skin-plating to which they are secured, must be properly worked on the plating seams.

TABLE* LXIII.

MINIMUM THICKNESS OF OUTSIDE PLATING AND FLAT PLATE KEEL.

Numerals.†	17 Years Class. Thickness in Lbs., Per Square Foot.									
	Sheer Strake.		Sheer to Bilge.		Bilge and Bottom.		Garboards.		Flat Plate Keel.	
	Half Length Amidships.	Ends.	Half Length Amidships.	Ends.	Half Length Amidships.	Ends.	Half Length Amidships.	Ends.	Half Length Amidships.	Ends.
2,000 and under 3,500	12	10	10	9	11	10	12	12	16	12
5,000 " 6,500	16	14	13	11	14	12	16	15	18	14
8,000 " 10,000	19	16	15	12	16	13	18	16	22	18
14,000 " 16,500	24	19	17	14	19	16	21	19	28	22
19,000 " 22,000	27	20	19	15	21	17	23	21	32	24
30,000 " 36,000	29	22	22	18	24	20	26	24	35	27
42,000 " 48,000	30	24	24	19	26	21	28	25	37	28
56,000 " 64,000	31	25	26	21	28	22	30	27	39	29
72,000 " 80,000	33	26	28	22	30	24	32	29	41	31
100,000 " 110,000	36	27	30	24	33	26	35	30	44	32

NOTE.—The following to be the minimum width of main sheer strake for $\frac{3}{5}$ length amidships for vessels of all grades. Numeral under 10,000, 33 inches; numeral 10,000 and under 16,500, 36 inches; numeral 16,500 and under 22,000, 40 inches; numeral 22,000 and above, 45 inches.

The minimum width of garboards and flat-plate keels for $\frac{3}{5}$ length amidships for vessels of all grades to be as follows: numeral under 10,000, 30 inches; numeral 10,000 and under 16,500, 33 inches; numeral 16,500 and above, 36 inches.

* The table is reproduced in part and for the 17-year class only.

† The numeral = (Depth + $\frac{1}{2}$ breadth + $\frac{1}{2}$ girth) in ft. $\times$ length, in ft.

Butt-straps :

The widths for single-, double-, and treble-riveted butt-straps, suited to different series of rivets, are specified in Table LXIV. * * * Butt straps must in no case be less than 2 lbs. thicker than the plates to which they are secured. Treble-riveted butt straps must, in all cases, be at least 4 lbs. thicker than the plates to which they are secured.

Table LXIV.

For Diameter of Rivets, Breadth of Laps, Lapped Butts, Width of Butt Straps and Breadth of Edge Strips on Plate Seams.

Thickness of Plates in lbs. weight per square foot.	10	$12\frac{1}{2}$	15	$17\frac{1}{2}$	20	$22\frac{1}{2}$	25	$27\frac{1}{2}$	30	$32\frac{1}{2}$	35	$37\frac{1}{2}$	40
Diameter of Rivets in sixteenths of an inch.	9	10	11	12	12	12	14	14	16	16	18	18	20
Size in inches of Countersink for Rivets of Plating.	1	1	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{6}{16}$	$1\frac{6}{16}$	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{12}{16}$	$1\frac{12}{16}$	$1\frac{12}{16}$
Breadth of Laps, in inches, for Single Riveting.	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$							
Breadth of Laps, in inches, for Double Riveting.	$3\frac{1}{2}$	$3\frac{3}{4}$	4	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$5\frac{1}{4}$	$5\frac{1}{4}$	6	6	7	7	$7\frac{1}{2}$
Width of Butt Straps, in inches, Double Riveted.	7	$7\frac{3}{4}$	$8\frac{1}{2}$	$9\frac{1}{2}$	$9\frac{1}{2}$	$9\frac{1}{2}$	11	11	$12\frac{1}{2}$	$12\frac{1}{2}$	14	14	$15\frac{1}{2}$
Width of Butt Straps, in inches, Treble Riveted.	$10\frac{1}{2}$	$11\frac{1}{2}$	$12\frac{3}{4}$	14	14	14	$16\frac{1}{2}$	$16\frac{1}{2}$	$18\frac{1}{2}$	$18\frac{1}{2}$	21	21	23
Breadth of Edge Strips for Plate Seams, in inches, for Single Riveting.	4	$4\frac{1}{2}$	$4\frac{3}{4}$	$5\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{4}$	$8\frac{1}{4}$	9
Breadth of Edge Strips for Plate Seams, in inches, for Double Riveting.	$7\frac{1}{2}$	8	$8\frac{3}{4}$	$9\frac{3}{4}$	$9\frac{3}{4}$	$9\frac{3}{4}$	11	11	13	13	$14\frac{1}{4}$	$14\frac{1}{4}$	$15\frac{3}{4}$
Breadth of Double-riveted Butt Laps.	$4\frac{1}{2}$	$4\frac{1}{2}$	5	5	5	6	6	6	6	6			
Breadth of Treble-riveted Butt Laps.					$7\frac{1}{2}$	$7\frac{1}{2}$	9	9	9	9	$10\frac{1}{2}$	$10\frac{1}{2}$	$10\frac{1}{2}$

The number and thickness of butt-straps varies with the vessel's numeral for plating. The following specification applies to the hull whose midship section is shown in Fig. 98.

Vessels whose numeral is 30,000 and under 48,000 to have the butts of sheer strake, 2 strakes of plating at bilge and upper deck stringer-plate secured with treble-riveted butt straps for $\frac{3}{4}$ the vessel's length amidships. The butt straps of bilge and shear-strakes, also deck stringer-plate if it is under 54 ins. wide, to be 7 lbs. thicker than the plate to which they are secured. The back row of rivets in foregoing butt straps to be spaced similar to the other rows. In addition to above, the remaining skin-plates are to be secured at their butts with treble-riveted butt straps 7 lbs. thicker than the plates to which they are secured for $\frac{2}{3}$ the vessel's length amidships. If any of the foregoing skin plates exceeds 54 ins. in width, the butts of same are to have butt straps 10 lbs. thicker than the plates to which they are secured. * * * When the vessel's numeral is, or exceeds, 16,500, the lapped butts of outside plating are to be treble-riveted throughout.

Rivets and Rivet-Work:

1. The diameters of rivets for the different thicknesses of plates and angle bars are specified in Table LXIV. 2. The longitudinal laps of skin plating, except main sheer strake, worked in, out- and in-strakes, and which is sixteen pounds and above in thickness, to be double-riveted. Main sheer strakes fourteen pounds and above to have the lap at their lower edge double-riveted. 3. The longitudinal seams of skin plating that is worked flush, and which is twenty pounds and above in thickness, to be secured with edge strips having two rows of rivets on each side of seam. 4. The longitudinal laps of skin-plating which is under the above specified thicknesses, for double-riveting, to be single-riveted. 5. All double-riveting in longitudinal laps and edges to be chain fashion, the distance between the rows in lap riveting to be not less than two and three quarter times, nor more than three times the diameter of rivet, from centre to centre of rivet and the laps are to be not less in width than six times the diameter of rivet. Double riveted edge strips to be the width specified in Table LXIV., and the spacing of rivets between rows to be similar to that hereafter specified for double-riveted butt straps. 6. Single-riveted laps to be, in width, three and a half times the diameter of rivet. Single-riveted edge strips to be the width specified in Table LXIV. 7. Longitudinally the distance between rivets in laps and edges, of skin plating, and the laps and seams of all plating required to be calked watertight, to be, from centre to centre, four times the diameter of rivet, providing the plating does not exceed twenty pounds in thickness; if the plating exceeds twenty pounds in thickness, the distance may be four and a half times the diameter of rivet. 8. The rivets in all butts — except the third row of a butt which is treble-riveted — are to be spaced apart, from centre to centre, three and a half times the diameter of rivet; and the distance between rows of butt rivets to be from two and a half to three diameters of rivet, from centre to centre of row. 9. The rivets in the third row of a butt, which is treble-riveted, may be seven diameters of rivets apart, from centre to centre, except otherwise specified. 10. The spacing of the rivets which secure the frames to skin plating, and to floor plates, to be, from centre to centre, not more than seven and a half times the diameter of rivet, except frames having watertight bulkheads secured to them, in which the spacing, from centre to centre, must not exceed five times the diameter of rivet. * * * 15. When the thickness of skin-plating amidships demands double-riveted laps or edges, the same is to be continued right fore and aft. 16. The diameter of rivets for securing plates, or plates and angle bars, of different thicknesses, to each other to be regulated by the thicker of the two. 17. When three or more thicknesses are riveted together, the thickest of the parts is to regulate the diameter of rivet. * * * 20. Rivet holes are to be fairly and regularly pitched, and must in no case be nearer the edge of a plate or angle bar than their diameter. 21. It is recommended that all rivet holes be punched one sixteenth of an inch smaller than the diameter of rivet to be used, and the holes be reamed to the size of rivet after the parts are in place. Any structure, or parts, where more than two thicknesses of material are riveted together, and all longitudinals, floors or brackets in double bottom, also keelsons and stringers in holds, to have their rivet holes punched one sixteenth less than the size of rivet to be used, and the holes reamed to size of rivet after the parts are in place. 22. Rivet holes are to be punched from the faying surfaces of the different parts, and great care must be taken to have them, in the different parts joined, truly opposite each other. When holes, in the parts joined, are not truly opposite each other, heavy drifting must not be resorted to; the holes must be reamed or drilled fair and a larger size rivet used. 23. The rivet holes in frames opposite skin-plate laps or edges to be drilled after the plates are fitted in place. * * * 25. Rivets in skin plating * * * to have their necks beveled under the rivet-heads so as to fill the countersink made in punching. 26. Rivet-heads should not be thicker than $\frac{2}{3}$ the diameter of rivet. The countersinking of all plates and angle-bars

to be made by drill and the countersink to extend right tnrough the plate or angle-bar.
27. Each rivet to fill its hole, the heads of those for skin plating to be close laid up and the rivets outside finished flush and fair, except in keel, stem, and stern-post, where they must be slightly convex.

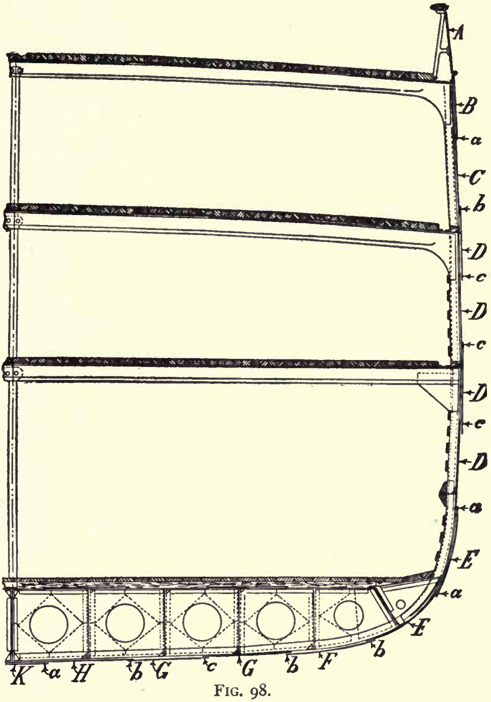


FIG. 98.

Fig. 98 is a midship section of a typical steel, 3-deck, screw steamer of the 17-year class. The principal data are: Length, 415 ft.; breadth, 48 ft.; depth, 32 ft.; $\frac{1}{2}$ girth to second deck, 43.5 ft.; $\frac{1}{2}$ breadth moulded, 24 ft.; depth to upper deck, 32 ft.; numeral for outside plating $= (43.5 + 24 + 32)415 = 41,292$.

Longitudinal Seams.

a. Lap, 6 in. wide, 1 in. rivets;
b. Lap, $5\frac{1}{4}$ in. wide, $\frac{7}{8}$ in. rivets;
c. Lap, $4\frac{1}{2}$ in. wide, $\frac{3}{4}$ in. rivets.

PLATING AND TRANSVERSE SEAMS.

Plating.	Thickness in lbs. per sq. ft.			Seam.
	½ length amidships.	At ends.	Throughout.	Treble-riveted butt for.
A. Bulwarks.			10	
B. Sheer strake.	33	23		¾ length.
C.	25	18		⅔ length amidships.
D.	23	18		⅔ length amidships.
E.	30	20		¾ length amidships.
F.	28	20		⅔ length amidships.
G.	23	20		⅔ length amidships.
H. Garboards.			25	⅔ length amidships.
K. Keel, outer plate.	34	28		Throughout.
Keel, inner plate.	25	extends ⅗ length amid.		Throughout.

CHAPTER V.

KEYED JOINTS; PIN-JOINTS.

The term "key" is applied to two forms of removable fastenings: The key proper, which is splined in a shaft to prevent relative rotation and sometimes axial movement of an attached member, as a pulley or gear-wheel; and the "through-key" or "cotter" which joins parts subjected to tensile or compressive stress or to both, as the sections of a pump-rod, the strap and body of a connecting rod, etc. The key proper is purely a locking device designed to resist shearing stress on the sectional area formed by its breadth and length; the cotter not only unites the parts, but, if of suitable length, gives, through its taper, a limited range of axial adjustment, while it withstands shearing at two transverse sections, each the product of its breadth and depth. Both forms are made generally of steel, although wrought iron finds infrequent use.

54. Forms of Keys.

Keys for shafting may be classified as: *Sunk keys*, *i. e.*, those which are fitted in key-seats cut in the shaft and in the attached hub; *Friction keys*, for which the key-seat in the shaft is omitted and which drive, or are driven by, the latter through friction only; and *Keys on the Flat* which, in their action, are intermediate between the two former classes.

1. SUNK KEYS are almost universally of the *square* or *flat* forms, shown in Figs. 99 and 100, respectively.

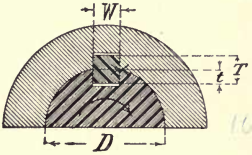


FIG. 99.

Square Keys prevent relative rotation only. They are approximately square in section with opposite sides parallel; the width, W, is slightly less than the depth, T, and the key is sunk in the shaft a little more than $\frac{1}{2}$ T; the key bears only on the sides of the key-seats in shaft and hub, there being usually a slight clearance at top and bottom. Such a key will not secure the attached hub against axial movement. The latter must be prevented by set-screws pass-

ing through the hub and bearing on the key; by making the hub a shrinkage or forced fit on the shaft; by splitting the hub, boring it for a pressure fit, and drawing the split together with bolts; by using loose collars with set-screws on the shaft at the ends of the hub; or, finally, if the hub be keyed at the extremity of the shaft, by threading a nut on the latter. On the other hand, the square key drives practically through its resistance to shearing stress on a longitudinal section, and, therefore, exerts no bursting pressure upon the hub and has no tendency to force the latter into eccentricity with the shaft. Hence, while its liability to tipping in its seat unfits it — unless secured by screws or dowels — for heavy loads, it is suitable for machine tools or other work in which accurate concentricity is required or in which the parts may be disconnected frequently.

The *Flat Key* (Fig. 100) locks both axially and circumferentially. Its section is rectangular and its sides are parallel, but its top and bottom, while plane, are inclined toward each other to form a wedge. The key is fitted accurately on all four surfaces. When driven home, the compression and elasticity of its metal and that of the hub, lock the latter effectually against motion in any direction. There are, however, a bursting pressure upon the hub and a tendency to spring the latter out of truth, both with the axis and with a plane normal to it. This key, as Mr. John Richards has pointed out, drives as a diagonal strut rather than by pressure normal to its face. If the angle, a, Fig. 100, be made about 30°, fair proportions will be obtained with a reasonably low value for the magnitude of the bursting element of the driving force. The taper is usually $\frac{1}{8}$-inch per foot. The key is suitable for heavy or light work in which slight inaccuracy in adjustment is not material.

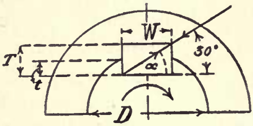


FIG. 100.

The *Feather Key* is a square key fitted for relative axial movement of the connected parts, as in a clutch-coupling. The key is secured in a key-seat in either the shaft or hub and the other keyway is made a working fit. The necessary surface to prevent wear from sliding movement may be had by increasing the length and to some extent the depth of the key. The latter is fastened to the seat either by countersunk screws, or by dove-tailed ends, or, in the case

of a hub, by gib (hooked) heads at the extremities of the feather. When the required surface warrants their use, two feather-keys set diametrically apart are better than one in equalizing the strain.

The *Woodruff Key* * (Fig. 101) may be of either the "square" or "flat" types. It has one-milled key-seat with parallel sides, but of circular outline at the bottom and hence of varying depth; the other key-seat is of the usual form. The maximum depth of the circular key-seat is considerably greater than that of the ordinary type and the shaft is, at that point, correspondingly weaker. On the other hand, the key is so firmly inset that it cannot possibly tip in its seat as the usual key may; and, further, the circular key will rotate in its seat until accurate adjustment with an angular spline is obtained, while the ordinary taper key may bear at one point only, if not well fitted. In order to avoid cutting too deeply into the shaft in securing a long hub, two or more "square" Woodruff keys may be inset in axial alignment with each other so as to engage the same key-seat in the hub.

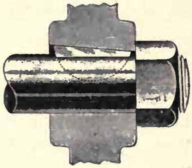


FIG. 101.

"*Quartering*" *Keys.* — With large shafts, especially when the hub to be secured is a loose fit, it is better to use two keys set 90° apart on the shaft, since the second key will oppose the hub's tendency to rock on the single key as a pivot. If two sunk keys be thus used, the width, *W*, of each need be but one half that required for a single key while the thickness, *T*, will also be less. In some cases, a sunk key is used to do the driving while a saddle-key or key on the flat, set quartering, steadies the hub and gives a rigid joint.

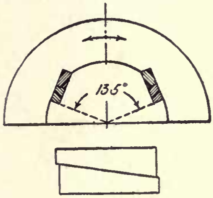


FIG. 102.

The *Peters System* of semi-sunk keys is shown in Fig. 102. It is suitable especially for fastening members having a reciprocating motion, either rotary or rectilinear, as, for example, a rock-shaft arm. There are two pairs of keys set preferably 135° apart. The keys of each pair have each one parallel and one tapered side. The latter engage, while the parallel

* The Whitney Manufacturing Co., Hartford, Conn.

sides abut against those of parallel-sided key-seats. The seats are normal to a radial and are partly in both shaft and hub, so that, for motion in either direction, the system supplies a radial driving face. The tapered sides enable the keys, when driven home, to make a rigid joint.

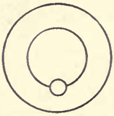


FIG. 103.

Pin-Keys, Fig. 103, may be used when the hub to be secured is on the end of the shaft. A cylindrical or taper hole is drilled and reamed at the shaft circumference, parallel to the axis, and one half each in shaft and hub. Into this hole a closely fitting cylindrical or taper-pin is driven which thus forms a sunk key. The method is accurate and cheap but is used only with light work.

2. FRICTION KEYS. — With this form no key-seat is cut in the shaft, the holding power of the key being due to friction only.

The *Saddle Key* is shown in Fig. 104. The sides are parallel, the top tapered, and the bottom concave, to fit the shaft. When the key is driven home, the friction causes it to grip the shaft. Its driving power is small and the principal uses of the key are to prevent rocking, when set quartering with a sunk key, and in temporary service, as in setting an eccentric. Locomotive eccentrics are sometimes secured permanently by two saddle-keys, fitted 90° apart, whose curved faces have longitudinal grooves or teeth which cut into the shaft when the keys are driven home.

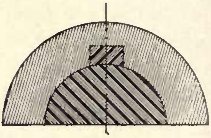


FIG. 104.

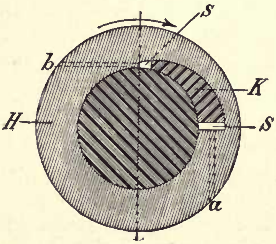


FIG. 105.

The *Kernaul Key*,* Fig. 105, drives only in one direction. The key, *K*, is approximately a segment somewhat less than 90° in extent, the inner face of which is curved to the radius of the shaft, the outer to that of an eccentric slot, *S*, formed in the hub, *H*. The inner surface of the key is left rough, the outer being finished and smooth. Hence, when the hub is rotated in the direction of

* Reuleaux's "Constructor," Suplee translation, 1895, p. 49.

the arrow, it slides over the key until the latter grips and revolves the shaft. A set-screw at *a* is used to set up the key and a similar one at *b* to loosen it.

The *Blanton Fastening*,* while not a friction-key, is, to some extent, a modification of the Kernaul principle. If, in Fig. 105, the key, *K*, be fixed on the shaft, it is evident that, with the circumferential clearance shown in the slot, *S*, the hub, *H*, will drive the shaft when rotated in the direction of the arrow; but, when the direction of rotation is reversed, the hub will become loose with limited angular play, so that it may be slipped along the shaft and removed readily. This is essentially the principle of the Blanton fastening which is applicable especially to the lifting cams of ore stamp-mills, largely because of the ease with which the cams may be disconnected and others substituted. The surface of the shaft is formed in a series of corrugations corresponding with a series of keys, *K*, and the hub is slotted to fit, with clearance at the ends of the slots, which ends are inclined and not radial, as in Fig. 105.

Roller Keys. — If, in Fig. 105, there be substituted for the key a hardened steel cylindrical roller whose diameter is a little less than the maximum radial width of the slot, *S*, it is obvious that a slight rotation of the hub will cause the roller to bind and thus drive the shaft. The connection is, like the Blanton, readily disengaged but is suitable for light work only. In fitting it, the ends of the hub are bored concentrically with the shaft and to the diameter of the latter, while the central recess for the key is circular but eccentric with the shaft.

The *Cone-Key* depends wholly upon friction for its driving power. The hub of the pulley or other member to be secured, is bored centrally with a tapered hole whose least diameter is greater than that of the shaft. A cast-iron bushing, bored to fit the shaft and turned to the taper of the hub-bore, is then split longitudinally into three equal parts, giving thus three saddle-keys, each nearly 120° long circumferentially. These keys, forced between shaft and hub, hold and drive the latter. The diameter of the shaft may be varied, within limits, for the same hub, by using a cone-key of the proper bore. Perhaps the most effective application of this principle is the Sellers Double Cone Coupling for shafts, patented and manufactured originally by William Sellers and

* F. R. Jones: "Machine Design," 1899, Part II., p. 205.

Company. As shown in Fig. 106, each shaft, as *A*, of the two to be connected is surrounded by a hollow cone, *B*, split only in one

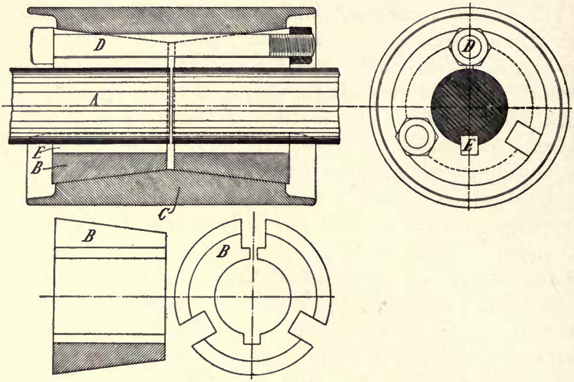


FIG. 106.

place. The cone is bored to fit the shaft and turned to a taper corresponding with the bore of one end of the encircling shell or "muff," *C*. The cone-bushings are bound to the shell and shaft through friction due to the axial stress upon three bolts, *D*, of square cross-section which lie in rectangular slots in both cones and draw the latter together and into the shell. As an additional precaution against slipping, each cone is attached positively to its shaft by a sunk key, *E*. The taper of the cones is about 1 : $7\frac{1}{2}$.

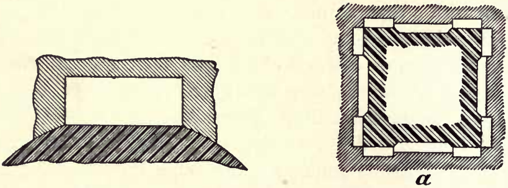


FIG. 107.

3. KEYS ON THE FLAT.—This type, Fig. 107, is, in driving power, intermediate between saddle and sunk keys, being recessed in the hub and bedded on a flat planed on the shaft. The upper

side of the key is tapered. Fastenings of this character were used formerly, as in Fig. 107*a*, in securing large hubs, as those of paddle-wheels, on square shafts. In such cases, the keys not only lock the hub in place, but may be used, within limits, to align it with the shaft.

55. Proportions of Keys.

The proportions of keys and key-seats have not been standardized and, in practice, show some variation.

1. General Proportions.—The following tables give the proportions recommended for general work by Mr. John Richards.* The notation refers to Figs. 99 and 100.

Table LXV.

Square (Straight) Keys.
(John Richards.)

D	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4
W	$\frac{5}{32}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{11}{32}$	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{17}{32}$	$\frac{9}{16}$	$\frac{11}{16}$
T	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$

Groove in shaft should be $\frac{9}{16}$ T in depth. Keys should not bear at top and bottom.

Table LXVI.

Flat (Taper) Keys.
(John Richards.)

D	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	7	8
W	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$
T	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	1

For shafts larger than those given in the table, there should be two or more keys, the width of which may be $\frac{1}{6}D$ while the depth may be obtained by making angle $a = 30°$.

Table LXVII.

Feather (Sliding) Keys.
(John Richards.)

D	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$
W	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
T	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$
L	3	$3\frac{1}{2}$	4	5	6	7	9	11	13	15

L = Maximum length. With feather fixed in hub, the shaft key-way should be a little the deeper.

* "Manual of Machine Construction," 1889, p. 57.

2. SHAFTING. — Professor Coleman Sellers, E. D., gives * the following proportions :

TABLE LXVIII.

KEYS (SQUARE) FOR SHAFTING.

(WILLIAM SELLERS & CO.)

Diameter of Shaft.	Size of Key.
$1\frac{1}{4}''-1\frac{7}{16}''-1\frac{11}{16}''$	$\frac{5}{16}''$ x $\frac{3}{8}''$
$1\frac{15}{16}-2\frac{3}{16}$	$\frac{7}{16}$ x $\frac{1}{2}$
$2\frac{7}{16}$	$\frac{9}{16}$ x $\frac{5}{8}$
$2\frac{11}{16}-2\frac{15}{16}-3\frac{3}{16}-3\frac{7}{16}$	$\frac{11}{16}$ x $\frac{3}{4}$
$3\frac{15}{16}-4\frac{7}{16}-4\frac{15}{16}$	$\frac{13}{16}$ x $\frac{7}{8}$
$5\frac{7}{16}-5\frac{15}{16}-6\frac{7}{16}$	$\frac{15}{16}$ x 1
$6\frac{15}{16}-7\frac{7}{16}-7\frac{15}{16}-8\frac{7}{16}-8\frac{15}{16}$	$1\frac{1}{16}$ x $1\frac{1}{8}$

Length of key-seat for coupling $= 1\frac{1}{2} \times$ nominal diameter of shaft.

3. MACHINE TOOLS. — The following tables are given herein through the courtesy of Messrs. William Sellers and Company and the Brown and Sharpe Manufacturing Company :

TABLE LXIX.

KEYS (SQUARE) AND KEY-SEATS FOR MACHINE TOOLS.

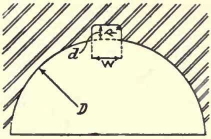
(WILLIAM SELLERS AND CO.)

Diameter of Shaft.	Size of Key.	Size of Key-Seat.
$\frac{1}{2}''$ and under.	$\frac{1}{16}''$ x $\frac{1}{16}''$	$\frac{1}{16}''$ x $\frac{1}{32}''$
Over $\frac{1}{2}''$	$\frac{1}{8}$ x $\frac{1}{8}$	$\frac{1}{8}$ x $\frac{1}{16}$
1 $''$ and $1\frac{3}{16}''$	$\frac{3}{16}$ x $\frac{3}{16}$	$\frac{3}{16}$ x $\frac{3}{32}$
$1\frac{1}{4}$ " $1\frac{7}{16}$	$\frac{1}{4}$ x $\frac{1}{4}$	$\frac{1}{4}$ x $\frac{1}{8}$
$1\frac{1}{2}$ " $1\frac{11}{16}$	$\frac{5}{16}$ x $\frac{5}{16}$	$\frac{5}{16}$ x $\frac{5}{32}$
$1\frac{3}{4}$ " $2\frac{3}{16}$	$\frac{7}{16}$ x $\frac{7}{16}$	$\frac{7}{16}$ x $\frac{7}{32}$
$2\frac{1}{4}$ " $2\frac{11}{16}$	$\frac{9}{16}$ x $\frac{9}{16}$	$\frac{9}{16}$ x $\frac{9}{32}$
$2\frac{3}{4}$ " $3\frac{13}{16}$	$\frac{11}{16}$ x $\frac{11}{16}$	$\frac{11}{16}$ x $\frac{11}{32}$
4 " $5\frac{7}{16}$	$\frac{13}{16}$ x $\frac{13}{16}$	$\frac{13}{16}$ x $\frac{13}{32}$
$5\frac{1}{2}$ " $6\frac{15}{16}$	$\frac{15}{16}$ x $\frac{15}{16}$	$\frac{15}{16}$ x $\frac{15}{32}$
7 " $8\frac{15}{16}$	$1\frac{1}{16}$ x $1\frac{1}{16}$	$1\frac{1}{16}$ x $\frac{17}{32}$
9 " $10\frac{15}{16}$	$1\frac{3}{16}$ x $1\frac{3}{16}$	$1\frac{3}{16}$ x $\frac{19}{32}$
11 " $12\frac{15}{16}$	$1\frac{5}{16}$ x $1\frac{5}{16}$	$1\frac{5}{16}$ x $\frac{21}{32}$
13 " $14\frac{15}{16}$	$1\frac{7}{16}$ x $1\frac{7}{16}$	$1\frac{7}{16}$ x $\frac{23}{32}$

* " The Stevens Indicator," IX., 2.

TABLE LXX.

KEY-WAYS FOR MILLING CUTTERS.
(BROWN AND SHARPE MANUFACTURING CO.)



Diameter (D) of Hole.	Width (W) of Key-way.	Depth (d) of Key-way.	Radius (R).
$\frac{3}{8}''$ to $\frac{9}{16}''$	$\frac{3}{32}''$	$\frac{3}{64}''$	.020″
$\frac{5}{8}$ " $\frac{7}{8}$	$\frac{1}{8}$	$\frac{1}{16}$	.030
$\frac{15}{16}$ " $1\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{64}$	.035
$1\frac{3}{16}$ " $1\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	.040
$1\frac{7}{16}$ " $1\frac{3}{4}$	$\frac{1}{4}$	$\frac{1}{8}$	.050
$1\frac{13}{16}$ " 2	$\frac{5}{16}$	$\frac{5}{32}$	.060
$2\frac{1}{16}$ " $2\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	.060
$2\frac{9}{16}$ " 3	$\frac{7}{16}$	$\frac{3}{16}$	.060

4. STATIONARY ENGINES. — The following table gives the practice of one of the leading builders in the United States with regard to the keys (square) for cranks and the flat (tapered) keys for the fly-wheels of stationary engines:

TABLE LXXI.

ENGINE KEYS.

Diam. of Shaft.	Width of Key.	Thickness.		Diam. of Shaft.	Width of Key.	Thickness.	
		Crank Key.	Wheel Key, Thin End.			Crank Key.	Wheel Key, Thin End.
$2\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{16}$	15	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{2}$
3	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	16	$2\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{5}{8}$
$3\frac{1}{2}$	$\frac{13}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	18	3	$2\frac{1}{2}$	$1\frac{3}{4}$
4	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	20	$3\frac{1}{4}$	$2\frac{3}{4}$	2
5	$1\frac{1}{16}$	$\frac{7}{8}$	$\frac{3}{4}$	22	$3\frac{5}{8}$	3	$2\frac{1}{8}$
6	$1\frac{3}{16}$	1	$\frac{3}{4}$	24	$3\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{1}{4}$
7	$1\frac{5}{16}$	$1\frac{1}{8}$	$\frac{7}{8}$	26	4	$3\frac{3}{8}$	$2\frac{3}{8}$
8	$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{7}{8}$	28	$4\frac{1}{4}$	$3\frac{1}{2}$	$2\frac{1}{2}$
9	$1\frac{5}{8}$	$1\frac{3}{8}$	1	30	$4\frac{1}{2}$	$3\frac{3}{4}$	$2\frac{3}{4}$
10	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{8}$	32	$4\frac{5}{8}$	4	$2\frac{7}{8}$
11	$1\frac{15}{16}$	$1\frac{5}{8}$	$1\frac{1}{8}$	34	$4\frac{3}{4}$	$4\frac{1}{8}$	$3\frac{1}{8}$
12	$2\frac{1}{16}$	$1\frac{3}{4}$	$1\frac{1}{4}$	36	5	$4\frac{1}{4}$	$3\frac{1}{4}$
13	$2\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{3}{8}$	38	$5\frac{1}{8}$	$4\frac{1}{2}$	$3\frac{3}{8}$
14	$2\frac{3}{8}$	2	$1\frac{3}{8}$	40	$5\frac{3}{8}$	$4\frac{5}{8}$	$3\frac{1}{2}$

The practice of this company is, with regard to:

Crank-Keys:

(*a*) No taper for crank-keys.
(*b*) Key to be $\frac{1}{2}$ in shaft and $\frac{1}{2}$ in hub, measured at *edge* of key-way.
(*c*) Use 2 keys, set 90° apart, for cranks bored $23\frac{3}{4}$ inches diameter and above.
(*d*) Use keys for nominal diameter of shaft.
(*e*) Keys in all counterbalanced cranks to be on the diameter passing through crank-pin, but on the opposite side of shaft from the pin.

Fly-Wheels:

(*a*) Taper of key, $\frac{1}{8}$-inch to 1 foot; tapered side in hub.
(*b*) *Thin* end of key to be $\frac{1}{2}$ in shaft and $\frac{1}{2}$ in hub, measured at *edge* of key-way.
(*c*) Use 2 keys, set 90° apart, for shaft 15 inches diameter and above.
(*d*) For sizes not given in table, use key for next smaller shaft.

5. MARINE ENGINES. — The following table is given herein through the courtesy of the Newport News Shipbuilding and Dry Dock Company. The notation (Fig. 100) is:

$D =$ diameter of shaft, ins.
$W =$ width of key and key-way, ins. $= \frac{3}{16}D + \frac{1}{8}''$.
$T =$ thickness of key $= \frac{3}{32}D + \frac{1}{8}''$.
$t =$ depth in shaft, measured at the side.
$T - t =$ depth in hub, measured at the side.
Taper $= \frac{1}{8}$ in. per foot.

TABLE LXXII.

KEYS (TAPERED) AND KEY-WAYS, MARINE ENGINES.
(NEWPORT NEWS SHIPBUILDING AND DRY DOCK CO.)

D	W	T	t	$T-t$	D	W	T	t	$T-t$
$\frac{1}{2}''$	$\frac{7}{32}''$	$\frac{3}{16}''$	$\frac{1}{16}''$	$\frac{1}{8}''$	$5''$	$1\frac{1}{16}''$	$\frac{5}{8}''$	$\frac{1}{4}''$	$\frac{3}{8}''$
$\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{8}$	$5\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{8}$
$\frac{3}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{1}{16}$	$\frac{1}{8}$	$5\frac{1}{2}$	$1\frac{3}{16}$	$\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{8}$
$\frac{7}{8}$	$\frac{9}{32}$	$\frac{7}{32}$	$\frac{3}{32}$	$\frac{1}{8}$	$5\frac{3}{4}$	$1\frac{3}{16}$	$\frac{11}{16}$	$\frac{5}{16}$	$\frac{3}{8}$
1	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{3}{32}$	$\frac{1}{8}$	6	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{5}{16}$	$\frac{3}{8}$
$1\frac{1}{8}$	$\frac{11}{32}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$6\frac{1}{4}$	$1\frac{5}{16}$	$\frac{3}{4}$	$\frac{5}{16}$	$\frac{7}{16}$
$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$6\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	$\frac{7}{16}$
$1\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$6\frac{3}{4}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	$\frac{7}{16}$
$1\frac{1}{2}$	$\frac{13}{32}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	7	$1\frac{7}{16}$	$\frac{13}{16}$	$\frac{5}{16}$	$\frac{1}{2}$
$1\frac{3}{4}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$7\frac{1}{4}$	$1\frac{1}{2}$	$\frac{13}{16}$	$\frac{5}{16}$	$\frac{1}{2}$
2	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$7\frac{1}{2}$	$1\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
$2\frac{1}{4}$	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$7\frac{3}{4}$	$1\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
$2\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{4}$	8	$1\frac{5}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
$2\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{4}$	$8\frac{1}{4}$	$1\frac{5}{8}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{9}{16}$
3	$\frac{11}{16}$	$\frac{7}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$8\frac{1}{2}$	$1\frac{3}{4}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{9}{16}$
$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$8\frac{3}{4}$	$1\frac{3}{4}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{9}{16}$
$3\frac{1}{2}$	$\frac{13}{16}$	$\frac{7}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	9	$1\frac{3}{4}$	1	$\frac{7}{16}$	$\frac{9}{16}$
$3\frac{3}{4}$	$\frac{13}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{5}{16}$	$9\frac{1}{4}$	$1\frac{7}{8}$	1	$\frac{7}{16}$	$\frac{9}{16}$
4	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{5}{16}$	$9\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$\frac{7}{16}$	$\frac{5}{8}$
$4\frac{1}{4}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$9\frac{3}{4}$	2	$1\frac{1}{16}$	$\frac{7}{16}$	$\frac{5}{8}$
$4\frac{1}{2}$	1	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	10	2	$1\frac{1}{16}$	$\frac{7}{16}$	$\frac{5}{8}$
$4\frac{3}{4}$	1	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{5}{16}$					

Propeller Keys.—The hub or boss of a screw-propeller is bored conically and fitted accurately to a corresponding taper on the after-end of the shaft, the latter being threaded for a nut which keeps the boss in place. The screw is driven by one or more longitudinal keys or feathers set in the tapered part of the shaft and fitting into suitable key-ways cut in the boss. These keys meet exceptionally severe service in rough weather when the ship is pitching and the position of the screw varies, in "racing," from partial to deep immersion. The screw-propellers for U. S. naval vessels are now made of manganese bronze or approved equivalent metal. With regard to the proportions of keys for naval propellers, Lieutenant-Commander F. H. Bailey, U. S. Navy, in charge of designs, Bureau of Steam Engineering, Navy Department, says:

"These keys are properly feathers since they are not usually driven, although this has been done in the case of some torpedo-boats. In our practice, the width of the key is about one and a half times its thickness, the latter being such that the side-pressure, calculated from the maximum turning moment on the shaft, shall not exceed about 25,000 lbs. per sq. in. on the propeller-hub. Thus, if a key is 2 ins. x 3 ins., bears for 30 ins. of its length, and is half in hub and half in shaft, the bearing surface would be 30 sq. ins. If the mean distance of the key from the centre of the shaft is 8 ins., the maximum turning moment on the shaft could be $30 \times 1 \times 25,000 \times 8 = 6,000,000$ inch pounds which maximum moment should be from 1.3 to 1.4 times the mean turning moment calculated from the horse-power and revolutions. Usually, we design the key so that the pressure on the key-way shall be about 22,000 lbs. per sq. in. If this pressure gives a key whose thickness is over $\frac{1}{8}$ of the shaft diameter, two keys set opposite are preferable."

56. Stresses on Keys.

Keys for shafting are subjected to shearing stress on the longitudinal cross-section and to crushing stress on the sides, or, when the key acts as a strut, in the direction of an approximate diagonal to the transverse section. As a general rule, it is better, in designing, to follow the empirical proportions given in the various tables which have been quoted, since keys with these dimensions, when well fitted and driven, seldom fail by either shearing or crushing. It is more probable that the shaft will be sheared or that the key will become loose in its seat. The latter action is soon fatal to the joint, since, through lost motion, vibration, and shock, the key, if not secured, will back out, or its sides or those of the key-seat will become so battered as to be useless. In work of an unusual nature or requiring especial care, keys may be designed or empir-

ical proportions tested by the application of the principles given below.

Shearing Stress on Key. — Let P be the load on the crank-pin or pulley-rim; R, the lever-arm of that load from shaft-centre; D, the diameter of the shaft; L, the length, and W, the width of the key; and S_s, the working unit shearing stress on the longitudinal cross-section. Then:

$$\text{Shearing resistance of key} = L \times W \times S_s;$$
$$\text{Moment of key-resistance} = L \cdot W \cdot S_s \times D/2;$$
$$\text{Moment of load} = P \times R.$$

Equating the moments:

$$W = \frac{2P \cdot R}{L \cdot D \cdot S_s}. \qquad (141)$$

Since the length, L, of the hub is known, the minimum value of W may be obtained from (141).

If the strength of the key against shearing is to be equal to that of the shaft in torsion, the width, W, may be found by equating the resisting moments of both. Let S_s' be the allowable shearing unit stress at the circumference of a solid cylindrical shaft, the polar modulus of the section being J/c. Then:

$$\text{Resisting moment of shaft} = S_s' \cdot \frac{J}{c} = S_s' \cdot \frac{\pi D^3}{16};$$

$$L \cdot W \cdot S_s \cdot \frac{D}{2} = S_s' \cdot \frac{\pi D^3}{16};$$

$$W = \frac{S_s'}{S_s} \cdot \frac{\pi}{8} \cdot \frac{D^2}{L}. \qquad (142)$$

For a hollow shaft of outer and inner diameters, D_1 and d, respectively, J/c becomes

$$\frac{\pi}{16} \cdot \frac{D_1^4 - d^4}{D_1}$$

and the leverage of the key is $D_1/2$.

Crushing Stress on Key. — The total resistance of a key to sidewise crushing is equal to the least area of the parts of the side inset in shaft or hub, multiplied by the working unit crushing

stress, S_c. The leverage of this resistance is, as before, $D/2$, approximately. Assume that the key is inset one half the depth, T. Then:

$$\text{Crushing resistance of key} = L \times T/2 \times S_c;$$
$$\text{Moment of key-resistance} = L \cdot T/2 \cdot S_c \times D/2.$$

Equating this moment with that of a solid, cylindrical shaft to torsion:

$$T = \frac{S_s'}{S_c} \cdot \frac{\pi}{4} \cdot \frac{D^2}{L}. \qquad (143)$$

Equating the values of L from (142) and (143):

$$T = 2 \cdot \frac{S_s}{S_c} \cdot W, \qquad (144)$$

which values apply to a key whose strength, in shearing and crushing, is the same and is equal to that of its shaft (solid, cylindrical) in torsion. If $S_c = 2S_s$, we have $T = W$, which is approximately true for square keys. For flat (taper) keys, which drive as a strut and are therefore relatively shallow, Richard's proportions (Fig. 100) give $T = W \tan 30°$. The crushing action on the sides is, other things equal, greater in square keys and feathers. The flat key is wedge-shaped, tends to drive on a diagonal to the cross-section, to tip in the seat, and thus to relieve the sides somewhat.

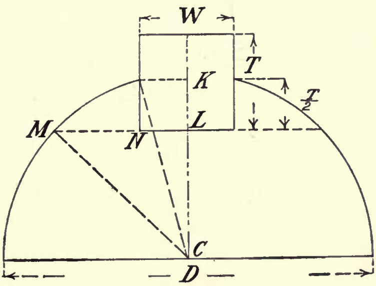


FIG. 108.

Shearing Stress on Shaft.—The load on a square key or feather acts to shear the shaft on the plane of the base of the key-seat, *i. e.*, in the direction, *M-N*, Fig. 108. Assume that the unit

shearing resistances of shaft and key are the same, that the key is sunk $\frac{1}{2}$ T in the shaft, and that $M\text{-}N = O\text{-}N = W$. Then:

$$T = 2K\text{-}L = 2(CK - CL).$$

Substituting the values of $C\text{-}K$ and $C\text{-}L$, we have, with sufficient approximation:

$$T = \frac{4W^2}{D} + \frac{10W^4}{D^3} + \text{etc.*}$$

Neglecting the last term and under the conditions given, the thickness of a key varies directly as the square of its breadth. Hence, since the shearing resistance varies as the breadth, the use of two or more keys in the place of one is attended, considering shearing stress only, by a reduction in the total metal used in keys and in the amount slotted out for key-ways.

The Grip of Friction Keys.—The holding power of these keys cannot be calculated with accuracy. The cone-key, Fig. 109, is driven home by a total maximum force, Q, which, through the expansion of the hub and the compression of the cone-bushing, produces the normal unit pressure, N, at the contact-surfaces of hub and key and the radial unit-pressure, P_0, at the joint between key and shaft. If θ be the half angle of the cone, the component of N which is normal to the axis, will be $P_1 = N \cos\theta$. Letting L = axial length of bearing, R_1 = mean radius of outer surface of cone, R_0 = radius of shaft, and μ and μ' = coefficients of friction, the resisting moment to circumferential slip will be, between:

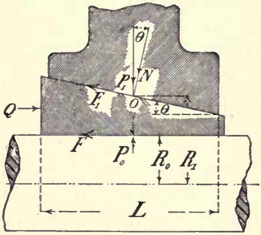


FIG. 109.

	Resistance.	*Moment.*
Shaft and bushing:	$2\pi R_0 L \times P_0\mu$;	$2\pi R_0 L P_0 \mu \times R_0$;
Bushing and hub:	$2\pi R_1 L \times P_1\mu'$;	$2\pi R_1 L P_1 \mu' \times R_1$;

which moments should be $\geqq P.R$, the turning moment on the shaft.

* Marks: "Relative Proportions of the Steam Engine," 1896, p. 97.

In these expressions, the surface removed in dividing the bushing is neglected. At the instant of driving home:

$$Q\,(\text{Max.}) = \sum N \sin\theta + F + F_1 \cos\theta,$$

in which F and F_1 are the total frictional resistances acting along the contact-surfaces. While Q may be measured, the values of μ and μ', as pointed out in §4, are uncertain in such cases. Again, the magnitude of P_0 is fixed by that of P_1 and the action of the intervening metal. Owing to the slots, the bushing cannot strictly be treated as either a thick cylinder or a thin band. In fair approximation, the grip may be estimated by considering the cone as a hollow cylinder of inner and outer radii, R_0 and R_1, respectively, subjected to the external pressure, P_1.

57. Through-Keys: Forms.

The through-key (cross-key, cotter) is simply a tapered cross-bar of rectangular or circular section driven through two members to be joined, as the sections of a pump-rod, a piston-rod and piston, a piston-rod and cross head, the strap and body of a connecting-rod, etc. The joint may be designed to resist tension only, as in foundation bolts; but is usually adapted for both tensile and compressive stresses. If the connected parts are movable axially and the key is sufficiently long; the latter gives means for longitudinal adjustment of the joint.

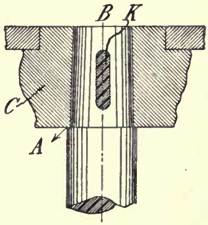


FIG. 110.

(*a*) *Cross-keyed Joints.* — Fig. 110 shows such a joint as used for connecting the piston and rod of a locomotive engine. The rod has a shoulder at A against which the piston is driven and on which it bears; its end has a sharp taper ($\frac{1}{8}$ in. in 4 ins.) from the shoulder to the extremity, B, and fits in a conical hole of the same taper in the piston, C; and the rod and piston are joined rigidly by a key, K, whose sides are parallel and whose top has a taper of $\frac{1}{4}$ in. in 12 ins. The sharp taper on the rod makes the parts readily detachable when the key is backed out.

In Fig. 111 * there are given two forms of a similar joint between the piston rods and crossheads of locomotive engines. In

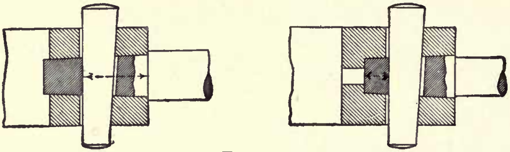


FIG. 111.

that to the left, the body of the rod is reduced for the taper and the crosshead is held by tension upon the extremity and weakest part of the rod, while, in the other type, the taper is made upon an enlargement of the rod and the latter bottoms in the fit, the joint being thus made by compressive stress. The latter method is much more secure than the former, in which the tension invites rupture.

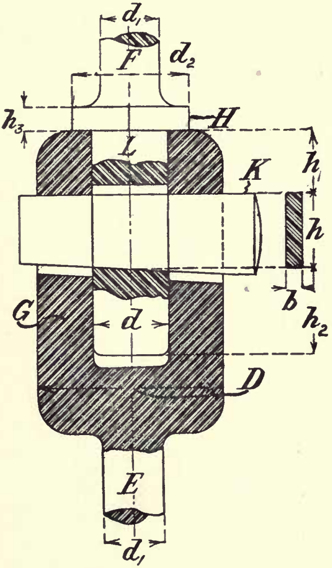


FIG. 112.

In Fig. 112 a similar connection between the sections of a pump-rod is shown. A socket, *G*, is formed on the lower section, *E*, in which the upper section, *F*, is recessed. Through the socket and the prolongation of the upper rod a key-way is slotted. The section, *F*, is held rigidly by the collar, *H*, formed on it and the key, *K*, passing through it and the socket.

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* *American Engineer and Railroad Journal*, January, 1899.

(*b*) *Gib and Key* (Fig. 113). — The brasses of the connecting-rod end shown in this figure are secured in place by the key, *K*,

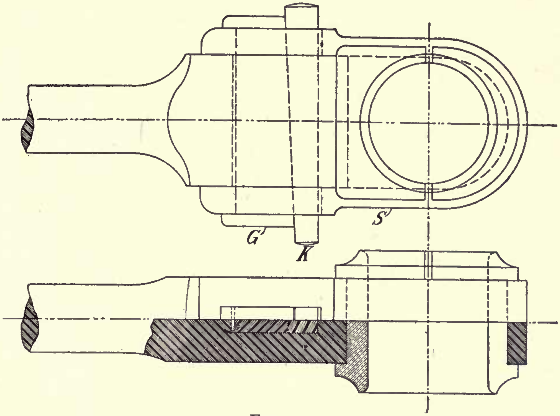


FIG. 113.

and gib, *G*, both of which pass through parallel-sided key-ways slotted through the strap, *S*, and the body of the rod. The abutting sides of gib and key have a taper of $\frac{1}{4}$ in. to $\frac{1}{2}$ in. in 12 ins.

As the key is driven home, the gib bears on the inner ends of the strap-slots and the key presses against the outer side of the key-way in the rod, thus drawing the brasses firmly together and against the body of the rod. The hooked ends of the gib overlap the strap and keep it from spreading. The location of the center of the journal is regulated by the liners between the brasses and by the position of the key. The latter gives, therefore, a limited range of adjustment as to the length of the rod.

In gib and key joints which are to be disconnected frequently or in which the pressure is excessive, the key may be tapered on both sides and pass between two gibs similar to *G*. This arrangement provides increased surface of a durable character for the key.

(*c*) *Bolted Strap-Ends.* — Fig. 114 and Table LXXIII. give the proportions of good practice in bolted strap-ends for connecting

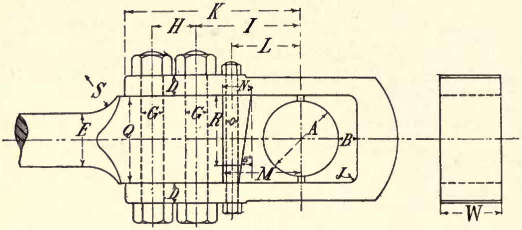


FIG. 114.

rods, a form which is more modern and in many respects more satisfactory than that shown in Fig. 113. In this type, the strap is secured by bolts, *G*, passing through it and the body of the rod, while the key, *N*, becomes a wedge simply which forces the brasses together and against the outer end of the strap. The taper (8 degrees) of the key is considerable and the position of the latter is regulated by the bolt, *O*, passing through it and through both forks of the strap.

(*d*) *Taper Pins.* — In light work, taper-pins are frequently used as cross-keys, as, for example, when driven into a diametrical hole, drilled and reamed to a corresponding taper, through the hub and shaft of a gear-wheel. The dimensions of the standard taper-pins made by the Morse Twist Drill and Machine Co., are:

Number.	0	1	2	3	4	5	6	7	8	9	10
Diameter at Large End, Inches.	.156	.172	.193	.219	.250	.289	.341	.409	.492	.591	.706
Approximate Fractional Sizes.	$\frac{5}{32}$	$\frac{11}{64}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{19}{64}$	$\frac{11}{32}$	$\frac{13}{32}$	$\frac{1}{2}$	$\frac{19}{32}$	$\frac{23}{32}$

The taper is $\frac{1}{4}$ in. per foot. The length ranges from $\frac{3}{4}$ in. for the No. 0 to 6 ins. for the No. 10, in increments of $\frac{1}{4}$ in.

(*e*) *Split Pins* (Table LXXVI.). — These pins may be used to prevent endwise motion in a nut or a pin-joint. They are circular in section, cylindrical or tapering in form, and are either split throughout, except at the head, or at the end only. When driven home, the pin is locked by spreading the split end.

TABLE LXXIII.

CONNECTING ROD ENDS.
(BOLTED STRAP.)

Diam. of Cylinder, Inches.	Lbs. Pressure on Piston 150 lbs. per sq. in.	*A*	*B*	*D*	*E*	*G*	*H*	*I*	*J*	*K*	*L*	*M*	*N*	*O*	*Q*	*R*	*S*	*W*	Diam. of Cylinder, Inches.
12	16,950	3	13/16	13/16	2 1/4	7/8	1 7/8	4 7/8	3/8	7 3/4	3	3 9/16	1 1/2	3/4	3 3/4	2 7/8	1 1/2	2 5/8	12
14	23,000	3 1/2	7/8	7/8	2 1/2	1	2 1/8	5 7/16	7/16	8 3/4	3 7/16	4	1 11/16	3/4	4 3/8	3 3/8	1 3/4	3	14
16	30,000	4	1	1	2 7/8	1 1/8	2 3/8	5 15/16	1/2	9 5/8	3 13/16	4 3/8	1 11/16	3/4	4 7/8	3 3/4	2	3 3/8	16
18	38,000	4 1/2	1 1/4	1 1/8	3 1/4	1 1/4	2 1/2	6 1/2	9/16	10 1/2	4 5/16	4 7/8	1 3/4	3/4	5 1/2	4 1/4	2 1/4	3 3/4	18
20	47,000	5	1 3/8	1 1/4	3 5/8	1 3/8	2 3/4	7 1/8	5/8	11 1/2	4 13/16	5 3/8	1 15/16	3/4	6	4 3/4	2 1/2	4 1/8	20
22	57,000	5 1/2	1 3/8	1 1/2	4	1 5/8	3 1/4	7 7/8	11/16	13 1/8	5 1/8	5 7/8	2 3/16	7/8	6 1/2	5 1/8	2 3/4	4 1/2	22
24	67,800	6	1 7/16	1 5/8	4 3/8	1 3/4	3 1/2	8 5/16	3/4	14	5 1/2	6 1/4	2 5/16	7/8	7	5 5/8	3	4 7/8	24
26	78,500	6 1/2	1 1/2	1 3/4	4 5/8	1 7/8	3 3/4	8 7/8	13/16	14 7/8	5 7/8	6 5/8	2 3/8	7/8	7 1/2	6	3 1/4	5 1/4	26
28	92,000	7	1 5/8	1 15/16	5	2	4	9 1/2	7/8	15 7/8	6 1/4	7 1/8	2 9/16	1	8	6 1/2	3 1/2	5 1/2	28
30	106,000	7 1/2	1 3/4	2 1/8	5 3/8	2 1/8	4 1/4	10 3/8	15/16	17 1/4	6 13/16	7 7/8	2 15/16	1 1/8	8 1/2	6 7/8	3 3/4	6	30
32	120,600	8	1 7/8	2 1/4	5 3/4	2 1/4	4 1/2	10 3/4	1	18	7 3/16	8 1/4	3	1 1/8	9	7 1/4	4	6 3/8	32
34	136,000	8 1/2	2	2 5/16	6 1/8	2 3/8	4 3/4	11 1/2	1 1/16	19 1/8	7 11/16	8 3/4	3 1/8	1 1/8	9 5/8	7 7/8	4 1/4	6 7/8	34
36	150,000	9	2 1/8	2 3/8	6 1/2	2 1/2	5	12 1/8	1 1/8	20 1/8	8 1/8	9 1/4	3 5/16	1 1/4	10 1/8	8 3/8	4 1/2	7 1/4	36
38	170,000	9 1/2	2 1/4	2 1/2	6 7/8	2 5/8	5 1/4	12 3/4	1 3/16	21 1/8	8 5/8	9 3/4	3 1/2	1 1/4	10 3/4	9	4 3/4	7 5/8	38
40	188,000	10	2 3/8	2 3/4	7 1/4	2 7/8	5 3/4	13 1/2	1 1/4	22 3/4	9 1/16	10 1/4	3 5/8	1 3/8	11 1/4	9 1/2	5	8	40
42	207,500	10 1/2	2 1/2	2 7/8	7 1/2	3	6	14 1/8	1 5/16	23 3/4	9 9/16	10 7/8	3 15/16	1 1/2	11 7/8	10 1/8	5 1/4	8 1/2	42
44	227,500	11	2 5/8	3	7 7/8	3 1/8	6 1/4	14 3/4	1 3/8	24 3/4	10 1/16	11 3/8	4 1/8	1 1/2	12 1/2	10 3/4	5 1/2	9	44
46	250,000	11 1/2	2 3/4	3 1/8	8 1/4	3 1/4	6 1/2	15 1/4	1 7/16	25 5/8	10 5/16	11 3/4	4 1/8	1 1/2	13	11 1/4	5 3/4	9 1/2	46
48	271,000	12	2 7/8	3 1/4	8 5/8	3 3/8	6 3/4	15 3/4	1 1/2	26 1/2	10 13/16	12 1/8	4 3/16	1 1/2	13 1/2	11 3/4	6	10	48

58. Through-Keys: Stresses.

1. TENSILE, SHEARING, AND BEARING STRESSES. — Assume the joint, Fig. 112, to be stressed axially and alternately in opposite directions. Failure may occur by:

Tensile Stress:

(*a*) On sections, E or F; (*b*) on section, L, where reduced by the keyway; (*c*) on socket, G, where similarly reduced; (*d*) on the key, due to bending.

Shearing Stress:

(*e*) On the key at inner surface of socket, G; (*f*) on socket, G, above key; (*g*) on collar, H; (*h*) on section, L, below key.

Bearing Stress:

(*k*) On key; (*l*) on section, L; (*m*) on socket, G; (*n*) on collar, H.

Take S_t, $S_s = 0.8S_t$, and S_b as the permissible unit tensile, shearing, and bearing stresses, respectively. The exact manner in which the key is loaded, is unknown. Assume it to be a simple beam, uniformly loaded with total stress, P. Then for condition:

(*a*) $$P = \frac{\pi}{4} \cdot d_1^2 \cdot S_t;$$

(*b*) $$P = \left(\frac{\pi}{4} \cdot d^2 - b \cdot d\right) S_t;$$

(*c*) $$P = \left[\frac{\pi}{4}(D^2 - d^2) - (D - d)b\right] S_t;$$

(*d*) $$\frac{Pd}{8} = \frac{b \cdot h^2}{6} \cdot S_t \therefore P = \tfrac{4}{3} \cdot \frac{bh^2}{d} \cdot S_t;$$

(*e*) $$P = 2(b \cdot h \cdot S_s);$$

(*f*) $$P = 2(D - d)h_1 \cdot S_s;$$

(*g*) $$P = \pi d \cdot h_3 \cdot S_s;$$

(*h*) $$P = 2(d \cdot h_2 \cdot S_s);$$

(*k*) $$P = b \cdot d \cdot S_b;$$

(*l*) $$P = b \cdot d \cdot S_b;$$

(*m*) $$P = b(D - d)S_b;$$

(*n*) $$P = \frac{\pi}{4}(d_2^2 - d^2)S_b.$$

It is evident that the more important of these stresses are shown by (*b*), (*d*) and (*e*). Taking $S_s = 0.8S_t$ and equating (*d*) and (*e*):

$$\tfrac{4}{3} \cdot \frac{bh^2}{d} \cdot S_t = 1.6bh \cdot S_t;$$

$$h = 1.2d. \qquad (145)$$

Substituting in (*e*) and equating (*b*) and (*e*):

$$1.92bd \cdot S = \left(\frac{\pi}{4}d - b\right) d \cdot S_t;$$

$$b = 0.27d, \text{ say } 0.25d, \qquad (146)$$

a ratio which conforms with good practice. The value of D in terms of d may be found, for tensile stress, by equating (*b*) and (*c*) and making $b = 0.25d$. This value, however, is less than that for bearing pressure obtained by equating (*l*) and (*m*). From the latter equations:

$$D = 2d. \qquad (147)$$

Equating the tensile and bearing resistances, (*b*) and (*l*), respectively, of rod L:

$$\left(\frac{\pi d}{4} - b\right) d \cdot S_t = b \cdot d \cdot S_b;$$

$$\therefore \frac{S_b}{S_t} = 2.14, \qquad (148)$$

a ratio which is not excessive with good materials and fitting. The depths, h_1 and h_2, if calculated for shearing simply by (*f*) and (*h*), are less than is required by good practice. Their value is usually from d to $1.25d$, with wrought iron. The diameter d_2, of the collar, H, should be greater proportionately in small rods, since the fillets and rounding greatly reduce the bearing surface. Taking the unit bearing pressure upon the collar as $\tfrac{2}{3}S_b$, we have, from (*l*) and (*n*) with $b = d/4$:

$$b \cdot d \cdot S_b = \frac{\pi}{4}\left(d_2^2 - d^2\right) \tfrac{2}{3} S_b;$$

$$d_2 = 1.22d. \qquad (149)$$

Equating (*e*) and (*g*) and substituting the values of *b* and *h*:

$$2bh \cdot S_s = \pi d \cdot h_3 \cdot S_s;$$

$$h_3 = 0.2d, \text{ about,} \tag{150}$$

a height which, considering the fillet of the collar, is sufficient.

2. Driving Force on Key.—Let A and B, Fig. 115, be two members of the same material united by a through-key, C, one side of the latter having the angle of taper, θ. Take W as the axial load upon the joint and μ and φ as the coefficient of friction and angle of repose, respectively, of C and A or B. Then, *disregarding the friction between the members*, A *and* B, in:

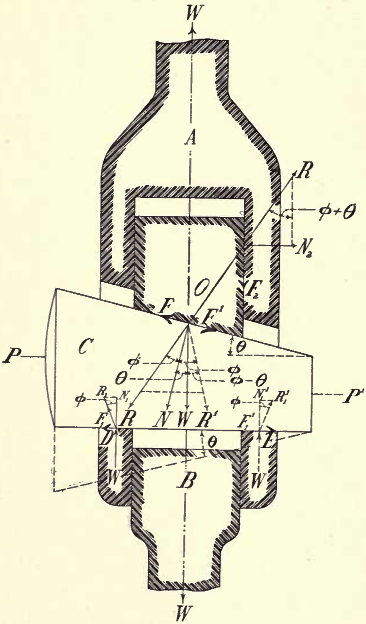


Fig. 115.

(*a*) *Driving home* the key, the latter is acted upon by the driving force, P, and the reactions, R and R_1, developed at the contact-surfaces by the load, W. The force, P, is opposed by the horizontal components of these reactions.

At the contact-surfaces of B and C, the load, W, taken as concentrated at O, produces the total normal pressure, N, which pressure, when the key moves, develops the force of friction, $F = \mu N = N \tan \varphi$. The reaction, R, is the resultant of N and F and is hence inclined from N and toward P by the angle φ, and from the vertical by the angle $\varphi + \theta$. The horizontal component of this reaction is $R \sin (\varphi + \theta)$.

The load, W, is divided between the two forks of member, A. Consider it as concentrated at D, at which point it produces a total normal pressure, N_1, and, when the key moves, a force of friction, $F_1 = \mu N_1 = N_1 \tan\varphi$, and a reaction, R_1, inclined toward P and from N_1 and the vertical by the angle φ. The horizontal component of this reaction is $R_1 \sin\varphi$. Hence:

$$P = R\sin(\varphi+\theta) + R_1 \sin\varphi; \qquad (151)$$

but, $R = W/\cos(\varphi+\theta)$ and $R_1 = N_1/\cos\varphi = W/\cos\varphi$

$$\therefore P = W[\tan(\varphi+\theta) + \tan\varphi]. \qquad (152)$$

(*b*) *In backing out* the key, consider the load as concentrated at O and E with regard to the members, B and A, respectively. The conditions are as before, excepting that the forces of friction, F' and F_1', act toward the backing force P'. Therefore, the reaction, R', is inclined from the vertical and toward P' by the angle $(\varphi-\theta)$ and the reaction, R_1', by the angle φ. Hence, as in (152):

$$P' = W[\tan(\varphi-\theta) + \tan\varphi]. \qquad (153)$$

(*c*) *Maximum Taper.* — If the angle θ is so great that the key, when driven home, is on the point of backing out, $P' = 0$. Hence,

$$\tan(\varphi-\theta) + \tan\varphi = 0$$

and

$$\theta = 2\varphi,$$

which is the limiting value for θ, when the key is not fitted with set-screws or other locking devices.

(*d*) *Friction of Members.* — The preceding equations neglect the friction between the members, A and B, and the value of W is therefore greater than the given force, P and P', would overcome in practice. Let W_f be the axial load, considering this friction. Then (Fig. 115) the reaction, R, will produce, between the contact-surfaces of A and B, a force of friction,

$$F_2 = \mu_2 N_2 = \mu_2 R \sin(\varphi+\theta),$$

which force will act downward in driving home. Hence, in raising B, the vertical loads to be overcome are:

$$W_f + \mu_2 R \sin(\varphi+\theta),$$

which quantity must be substituted for W in preceding equations. Hence, *considering the friction between the connected members:*

$$R = \frac{W_f + \mu_2 R \sin(\varphi + \theta)}{\cos(\varphi + \theta)} = \frac{W_f}{\cos(\varphi + \theta) - \mu_2 \sin(\varphi + \theta)};$$

$$R_1 = \frac{W_f + \mu_2 R \sin(\varphi + \theta)}{\cos \varphi}.$$

Substituting in (151):

$$P = [W_f + \mu_2 R \sin(\varphi + \theta)][\tan(\varphi + \theta) + \tan\varphi];$$

$$= W_f \cdot \frac{\tan(\varphi + \theta) + \tan\varphi}{1 - \mu_2 \tan(\varphi + \theta)}, \qquad (154)$$

in which μ_2 is the coefficient of friction for the metal of A and B. In finding the value of P' by a similar method, the force of friction is calculated from the normal pressure produced by R' and that force acts upward, in opposition to W_f, and hence is subtractive.

(*e*) *Double Taper.*—Assume that both sides of the key have the same angle of taper, θ, as shown by broken lines in Fig. 115. Then, R_1 and R_1' are equal to R and R', respectively, and, from (152) and (153):

$$P = 2W \tan(\varphi + \theta); \qquad (155)$$

$$P' = 2W \tan(\varphi - \theta), \qquad (156)$$

which equations neglect the friction of the connected members.

The limiting angle of taper at which this key, when driven home, is on the point of backing out, is found, as before, by making $P' = 0$. Hence:

$$\tan(\varphi - \theta) = 0 \therefore \theta = \varphi.$$

Under customary conditions, with slightly oily metals and without locking devices on the key, the latter will begin to back out when its taper reaches about $1\frac{1}{2}$ ins. per ft., *i. e.*, a ratio of 1 to 8. The taper for such keys is, in practice, much less, the ratio being usually 5 or 6 times this limit.

59. Pin-Joints.

Pin-joints, *i. e.*, those in which two or more members are united pivotally by a cylindrical pin meet frequent use.

1. Boiler Braces. — The joint may be as shown in Fig. 116 or the member, B, may be replaced by a lug or curved strap pass-

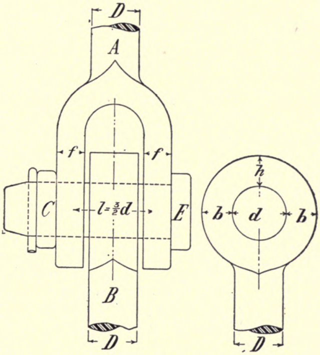


Fig. 116.

ing over the pin and riveted to the head-plate. In such a joint, if the parts be accurately fitted without lost motion between members, A and B, or between the pin-bearing and pin, the latter is subject only to double shear. The fit, however, is frequently loose; and, in any event, the pin and bearing will probably wear. Hence, the pin is subject frequently to both shearing and bending stresses and may fail as shown in Fig. 117.*

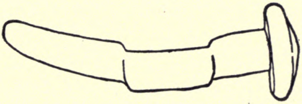


Fig. 117.

(*a*) *Brace-body.* — In general, let L be the length and B the breadth in ins. of the area supported by the brace and let p be the pressure per sq. in. upon that area. Then the total load on area and brace is:

$$W = L \cdot B \cdot p = \pi D^2/4 S_t, \tag{157}$$

in which S_t is the working unit tensile stress of the brace.

* *The Locomotive*, August, 1901.

(*b*) *Shearing Stress on Pin.* — Assuming accurate fitting throughout, the pin will be subject only to double shear. Then, by §45, 3, and taking S_s, the *mean* unit shearing stress, as $0.8S_t$:

$$W = 1.75 \cdot \pi d^2/4 \cdot S_s = 1.4 \cdot \pi d^2/4 \cdot S_t. \qquad (158)$$

Equating (157) and (158):

$$d = 0.84D.$$

(*c*) *Bending Stress on Pin.* — The distribution of the load upon the pin, in its bearings both in *B* and in the forks of *A*, is unknown. In any event, the pin acts as a supported beam of circular cross-section, as in Fig. 118. In extreme cases, the load, *W*, may be

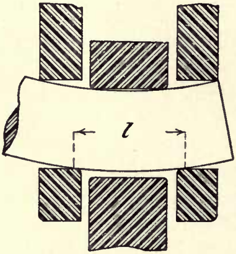


FIG. 118.

concentrated at the centre of the length, and the distance, l, between the supports may be the total width, *C-E*, Fig. 116, of the bearing. Assume $l = 1.5d$ and the load as concentrated, as above. Then, the maximum bending moment is:

$$M = \frac{Wl}{8} = S_t \cdot \frac{I}{c} = S_t \cdot \frac{\pi d^3}{32},$$

in which I/c is the modulus of the section. Then:

$$W = \tfrac{2}{3} \cdot \frac{\pi}{4} d^2 \cdot S_t. \qquad (159)$$

Taking the tensile stress due to bending as equal to that in direct tension and equating (157) and (159):

$$d = 1.2D.$$

For average practice, the diameter for shearing, as thus calculated, is too small and that for bending is large. Taking the mean:

$$d = D.$$

(*d*) *Sides and Crown of Eye.* — When the diameter of the pin is such that the eye may be tested to destruction, the latter fails by crushing at h, Fig. 116, and rupture at the two sections, b, the metal flowing so that the thickness of the eye is increased considerably at the inner limit of h and much decreased at those of b, where fracture appears first.

The section at b is subjected not only to tensile stress due to its share of the load, W, but also to an additional bending stress, owing to the distance between the line of application of the load and the centre of gravity of the section. With regard to the width of the crown at h, the problem is, in general, one of indentation (p. 184) and the stresses (§ 45, 46) resemble somewhat those in a thick, hollow cylinder under internal fluid pressure.* The case is also similar generally to that of the margin of a riveted joint, in which, for ample strength $E = 1.5d$ (84), *i. e.*, the distance from edge of hole to edge of sheet is d. In practice the periphery of the eye is concentric with the hole and $h = b = 0.5d$ to $0.75d$.

(*e*) *Member A.* — The thickness, f, of the forks should be proportioned for two thirds of the load to allow for inaccurate fitting and irregular distribution. Generally, $f = 0.66d$ to $0.75d$.

In good work, bosses for planing are formed on each side of the eye and where the head and washer of the pin fit, with a consequent increase of thickness at those points.

(*f*) *Tests.* — In 1879, Chief Engineers Sprague and Tower, U. S. Navy, made exhaustive experiments upon boiler braces. Their recommendations † are :

"The following is submitted for the proportions (with sufficient excess in the eye for wear, etc.) of the ends of boiler braces made in the manner specified. In the same bar, the section across the eye must be increased with each material increase of the diameter of the pin. When the brace is round and the thickness of the eye and the diameter of the bar are equal, let $x =$ areas.

"For ends made by drawing out the bar, bending it around and welding : $x =$ width of bar and the diameter of iron pin, $\frac{11}{17} x =$ diameter of steel pin, $\frac{8}{5} x =$ breadth of (concentric) eye, thickness of eye to equal that of bar.

"For ends cut from flat bars, $x =$ width of bar and diameter of iron pin, $\frac{2}{3} x =$ diameter of steel pin, $\frac{3}{4} x =$ breadth across each side of eye, $\frac{7}{8} x =$ depth through crown of eye, thickness of eye $=$ that of bar.

"For ends upset, and forged solid, holes drilled, $x =$ area of bar and area of iron pin, $1.48 x =$ area of section across the eye, $.9 x =$ area through crown of eye."

* Cotterill : "Applied Mechanics," 1895, p. 368.

† "Report on Experiments to Ascertain Proportions for the Ends of Boiler Braces," Washington, 1880.

2. Structural Work. — Pin-joints are used for trusses and in the lateral system. One such joint is shown in Fig. 119, which

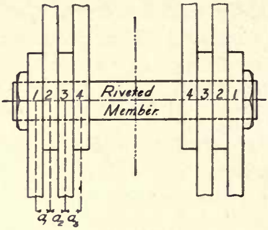


Fig. 119.

has a compound member in the centre with four sets of eye-bars in pairs, one bar of each pair being on each side of the centre. Large pins have a nut at each end; those of smaller diameter may have a head at one end and a split-pin, serving as a cotter, at the other. To allow for irregularities in thickness or fit, the "grip," or length of pin between the inner faces of the nuts, is increased, beyond the aggregate thickness of the connected members, by $\frac{1}{16}$ in. for each bar and $\frac{1}{4}$ in. for the riveted member as a rule.

The stresses on the bars and pin may be horizontal, vertical, or diagonal. Resolving the latter into horizontal and vertical components, the pin-stresses may be divided into four classes: Positive horizontal and negative horizontal stresses, acting toward the left and right, respectively; and positive vertical and negative vertical stresses, acting upward and downward, respectively. The pin, Fig. 119, may be considered as a beam, acted upon by various stresses, as above, each at a distance from the next stress corresponding with the thicknesses of the respective eye-bars and the allowance for irregularities. The diameter of the pin must be proportioned for bending, shearing, and bearing pressure.

(*a*) *Bending.* — If the maximum bending moment on the pin be known, the diameter of the latter for bending stress may be found from the fundamental formula:

$$M(\text{max.}) = S \cdot \frac{I}{c} = S \cdot \frac{\pi d^3}{32},$$

in which M is the maximum moment, S is the allowable working unit stress, and d is the diameter required.

To determine the maximum moment for any given manner of loading, the moment at the centre of each member must be found. each load being considered as concentrated at the centre of its respective bar. This moment will be the resultant of all preceding moments to the left. The principles of the resolution and composition of forces apply also to moments. Hence, the moment upon any section will be the resultant of the horizontal and vertical moments upon that section, *i. e.*, the square root of the sum of the squares of the latter moments.

Thus, assume in Fig. 119, stresses, P_1, P_2, P_3, P_4, upon the corresponding members, the lines of action being separated by the distances, a_1, a_2, a_3. Let P_1 be a positive horizontal stress; P_2, a diagonal stress with vertical and horizontal components, $+ P_2v$ and $- P_2h$, respectively; P_3, a diagonal stress with vertical and horizontal components, $- P_3v$ and $- P_3h$, respectively; and P_4 a negative vertical stress. Then at:

Member No. 2:

Horizontal Moment $= P_1 \times a_1 = H.M_2$;

Vertical " $= 0$;

Resultant " $= \sqrt{H.M_2^2 + 0} = H.M_2$.

Member No. 3:

Horizontal Moment $= P_1(a_1 + a_2) - P_2h \times a_2 = H.M_3$;

Vertical " $= P_2v \times a_2 = V.M_3$;

Resultant " $= \sqrt{H.M_3^2 + V.M_3^2}$.

Succeeding resultant moments may be calculated similarly. From a comparison of the results, the value and location of the maximum bending moment upon the pin may be found. Table LXXIV. gives the required diameters for various maximum moments and extreme fibre stresses per sq. in., as computed by the fundamental formula for bending moment. It will be observed that the calculations, as above, apply only to the pin before bending. When the latter occurs, the stress-leverages and maximum moment are reduced.

Table LXXIV.

Maximum Bending Moments on Pins.

Pin.		Moments in Inch Pounds for Fibre Strains per Square Inch of				
Diam.	Area.	15,000	18,000	20,000	22,000	25,000
1″	0.785	1,470	1,770	1,960	2,160	2,450
1¼	1.227	2,880	3,450	3,830	4,220	4,790
1½	1.767	4,970	5,960	6,630	7,290	8,280
1¾	2.405	7,890	9,470	10,500	11,570	13,200
2	3.142	11,800	14,100	15,700	17,280	19,600
2¼	3.976	16,800	20,100	22,400	24,600	28,000
2½	4.909	23,000	27,600	30,700	33,700	38,400
2¾	5.940	30,600	36,800	40,800	44,900	51,000
3	7.069	39,800	47,700	53,000	58,300	66,300
3¼	8.296	50,600	60,700	67,400	74,100	84,300
3½	9.621	63,100	75,800	84,200	92,600	105,200
3¾	11.045	77,700	93,200	103,500	113,900	129,400
4	12.566	94,200	113,100	125,700	138,200	157,100
4¼	14.186	113,000	135,700	150,700	165,800	188,400
4½	15.904	134,200	161,000	178,900	196,800	223,700
4¾	17.721	157,800	189,400	210,400	231,500	263,000
5	19.635	184,100	220,900	245,400	270,000	306,800
5¼	21.648	213,100	255,700	284,100	312,500	355,200
5½	23.758	245,000	294,000	326,700	359,300	408,300
5¾	25.967	280,000	335,900	373,300	410,600	466,600
6	28.274	318,100	381,700	424,100	466,500	530,200
6¼	30.680	359,500	431,400	479,400	527,300	599,200
6½	33.183	404,400	485,300	539,200	593,100	674,000
6¾	35.785	452,900	543,500	603,900	664,200	754,800
7	38.485	505,100	606,100	673,500	740,800	841,900
7¼	41.282	561,200	673,400	748,200	823,000	935,300
7½	44.179	621,300	745,500	828,400	911,200	1,035,400
7¾	47.173	685,500	822,600	914,000	1,005,300	1,142,500
8	50.265	754,000	904,800	1,005,300	1,105,800	1,256,600
8¼	53.456	826,900	992,300	1,102,500	1,212,800	1,378,200
8½	56.745	904,400	1,085,200	1,205,800	1,326,400	1,507,300
8¾	60.132	986,500	1,183,800	1,315,400	1,446,900	1,644,200
9	63.617	1,073,500	1,288,200	1,431,400	1,574,500	1,789,200
9¼	67.201	1,165,500	1,398,600	1,554,000	1,709,400	1,942,500
9½	70.882	1,262,600	1,515,100	1,683,400	1,851,800	2,104,300
9¾	74.662	1,364,900	1,637,900	1,819,900	2,001,900	2,274,900
10	78.540	1,472,600	1,767,100	1,963,500	2,159,900	2,454,400
10¼	82.520	1,585,900	1,903,000	2,114,500	2,325,900	2,643,100
10½	86.590	1,704,700	2,045,700	2,273,000	2,500,200	2,841,200
10¾	90.760	1,829,400	2,195,300	2,439,300	2,683,200	3,049,100
11	95.030	1,960,100	2,352,100	2,613,400	2,874,800	3,266,800
11¼	99.400	2,096,800	2,516,100	2,795,700	3,075,400	3,494,800
11½	103.870	2,239,700	2,687,600	2,986,300	3,284,800	3,732,800
11¾	108.430	2,388,900	2,866,600	2,185,200	3,503,700	3,981.500
12	113.100	2,544,700	3,053,600	3,392,900	3,732,190	4,241,200

(*b*) *Shearing.* — The vertical shear at any section of the pin is the algebraic sum of the vertical stresses to the left of that section. Similarly, the horizontal shear at the section considered is the algebraic sum of the horizontal stresses to the left of that section. Then, the resultant shear upon the section is the square root of

the sum of the squares of the vertical and horizontal shears, as above. Thus, the shears are at:

Member No. 3:

$$\text{Horizontal} = P_1 - P_2h = H.S_3;$$

$$\text{Vertical} \quad = 0 + P_2v = V.S_3;$$

$$\text{Resultant} = \sqrt{H.S_3^2 + V.S_3^2} = R.S_3.$$

While, in a cylindrical section, the maximum is $\frac{4}{3}$ the mean shearing stress (p. 182), it is usual to consider the shearing stress on pins as uniformly distributed over the cross-section. Again, since the bars are in pairs, the pin may be considered as under double shear. Hence, for one pair of bars:

$$S_s = 2R.S./1.75(\pi d^2/4),$$

in which $R.S.$ is the maximum resultant shear, as above, d is the diameter required to withstand that shear, and S_s is a unit working shearing stress which is low enough to permit, with safety, the excess of maximum over mean stress.

(*c*) *Bearing.* — The ranges of permissible bearing pressure and shearing stress have been given previously (p. 224).

(*d*) *Proportions.* — Table LXXV. gives the proportions of pins with Lomas nuts and Table LXXVI. of pins with cotters. The latter are used with pins of small diameters only. The former are preferable, since the nut is recessed and bears only on its periphery. This allows the body of the pin to enter it and enables it to be set up tightly when the aggregate thickness of the members with the allowances is not equal to the estimated grip of the pin.

(*e*) *Eyebars.*—The proportions of plain and adjustable eye-bars are given in Table LXXVII.

(*f*) *Specifications.* — The following extracts, referring to pin-joints, are taken from the specifications of the American Bridge Company for steel railroad bridges. The specifications for rivet, soft, and medium steel are given on page 219.

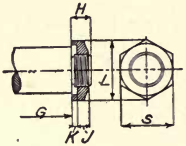
"Pins made of either of the above mentioned grades of steel shall, on specimen test-pieces cut from finished material, fill the requirements of the grade of steel from which they are rolled, excepting the elongation, which shall be decreased 5 per cent. from that specified.

"Pins up to 7 inches diameter shall be rolled.

TABLE LXXV.

PINS WITH LOMAS NUTS.

(AMERICAN BRIDGE CO.)



Diameter of Pin.	Pin.				Nut.						
	Diameter of Pin Hole.	Screw. Diam.	Screw. Length	Add to Grip.	Diam. of Rough Hole.	Short Diam. S	Long Diam. L	Weight of Nut.	H	J	K
2"	2 1/32"	1 1/2"	1 1/2"	1/4"	1 5/16"	3 1/4"	3 3/4"	2.5			
2 1/4	2 9/32	1 1/2	1 1/2	1/4	1 5/16	3 1/4	3 3/4	2.5			
2 1/2	2 17/32	2	1 1/2	1/4	1 13/16	3 3/4	4 5/16	2.5	1 1/2	1	3/8
2 3/4	2 25/32	2	1 1/2	1/4	1 13/16	3 3/4	4 5/16	2.5			
3	3 1/32	2 1/2	1 1/2	1/4	2 5/16	4 1/2	5 3/16	3			
3 1/4	3 9/32	2 1/2	1 1/2	1/4	2 5/16	4 1/2	5 3/16	3			
3 1/2	3 17/32	2 1/2	1 1/2	1/4	2 5/16	4 1/2	5 3/16	3			
3 3/4	3 25/32	3	1 7/8	1/2	2 13/16	5	5 3/4	5.5			
4	4 1/32	3	1 7/8	1/2	2 13/16	5	5 3/4	5.5			
4 1/4	4 9/32	3 1/2	1 7/8	1/2	3 5/16	5 3/4	6 5/8	7			
4 1/2	4 17/32	3 1/2	1 7/8	1/2	3 5/16	5 3/4	6 5/8	7			
4 3/4	4 25/32	3 1/2	1 7/8	1/2	3 5/16	5 3/4	6 5/8	7	1 7/8	1 1/4	1/2
5	5 1/32	4	1 7/8	1/2	3 13/16	6 1/2	7 1/2	8.5			
5 1/4	5 9/32	4	1 7/8	1/2	3 13/16	6 1/2	7 1/2	8.5			
5 1/2	5 17/32	4 1/2	1 7/8	1/2	4 5/16	7	8 1/8	11			
5 3/4	5 25/32	4 1/2	1 7/8	1/2	4 5/16	7	8 1/8	11			
6	6 1/32	4 1/2	1 7/8	1/2	4 5/16	7	8 1/8	11			
6 1/4	6 9/32	5	2 3/8	3/4	4 13/16	7 3/4	8 15/16	12			
6 1/2	6 17/32	5	2 3/8	3/4	4 13/16	7 3/4	8 15/16	12			
6 3/4	6 25/32	5 1/2	2 3/8	3/4	5 5/16	8 1/4	9 1/2	13.5			
7	7 1/32	5 1/2	2 3/8	3/4	5 5/16	8 1/4	9 1/2	13.5			
7 1/4	7 9/32	5 1/2	2 3/8	3/4	5 5/16	8 1/4	9 1/2	13.5			
7 1/2	7 17/32	5 1/2	2 3/8	3/4	5 5/16	8 1/4	9 1/2	13.5			
7 3/4	7 25/32	6	2 3/8	3/4	5 13/16	9	10 3/8	17			
8	8 1/32	6	2 3/8	3/4	5 13/16	9	10 3/8	17			
8 1/4	8 9/32	6	2 3/8	3/4	5 13/16	9	10 3/8	17	2 3/8	1 1/2	5/8
8 1/2	8 17/32	6	2 3/8	3/4	5 13/16	9 1/2	10 7/8				
8 3/4	8 25/32	6	2 3/8	3/4	5 13/16	10 1/2	12 1/8				
9	9 1/32	6	2 3/8	3/4	5 13/16	10 1/2	12 1/8				
9 1/4	9 9/32	6	2 3/8	3/4	5 13/16	10 1/2	12 1/8				
9 1/2	9 17/32										
9 3/4	9 25/32										
10	10 1/32										

NOTE.—To obtain grip G add $\frac{1}{16}$ for each bar, together with amount given in table.

"Pins exceeding 7 inches diameter shall be forged under a steel hammer striking a blow of at least 5 tons. The blooms to be used for this purpose shall have at least three times the sectional area of the finished pins.

"All pins shall be accurately turned to a gauge, and shall be straight and smooth.

"The clearance between pin and pin-hole shall be $\frac{1}{32}$ of an inch for all lateral pins; and for truss pins the clearance shall be $\frac{1}{50}$ of an inch for pins $3\frac{1}{2}$ inches in diameter, which amount shall be gradually increased to $\frac{1}{32}$ of an inch for pins 6 inches in diameter and over.

"All pins shall be supplied with steel pilot nuts, for use during erection.

"All pin-holes shall be reënforced by additional material when necessary, so as not to exceed the allowed pressure on the pins. These reënforcing plates must contain enough rivets to transfer the proportion of pressure which comes upon them, and at least one plate on each side shall extend not less than 6 inches beyond the edge of the tie plate.

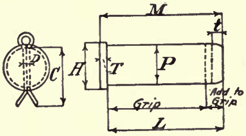
"Pin-holes shall be bored truly parallel with one another and at right angles to the axis of the member unless otherwise shown in drawings; and in pieces not adjustable for length, no variation of more than $\frac{1}{64}$ of an inch for every 20 feet will be allowed in the length between centres of pin-holes.

The permissible shearing strain and bearing pressure are given on page 226.

"The bending strain on the extreme fibre of pins shall not exceed 22,000 pounds per square inch for soft steel and 25,000 per square inch for medium steel, when centres of bearings of the strained members are taken as the points of application of the strains.

Table LXXVI.

Pins with Cotters.
(American Bridge Co.)



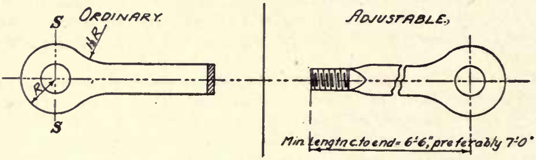
Pin.			Head.		Cotter.		Add to Grip.	
Diam of Pin. P	Diam of Pin-Hole.	Taper at End, t	Diam. H	Thickness, T	Length, C	Diam. D	For Length over All, M	For Length under Head, L
1"	$1\frac{1}{32}$"	$\frac{5}{16}$" x $\frac{1}{16}$"	$1\frac{1}{4}$"	$\frac{1}{4}$"	$1\frac{3}{4}$"	$\frac{1}{4}$"	$\frac{7}{8}$"	$\frac{5}{8}$"
$1\frac{1}{4}$	$1\frac{9}{32}$	$\frac{5}{16}$ x $\frac{1}{16}$	$1\frac{1}{2}$	$\frac{1}{4}$	2	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{5}{8}$
$1\frac{1}{2}$	$1\frac{17}{32}$	$\frac{7}{16}$ x $\frac{3}{32}$	$1\frac{3}{4}$	$\frac{1}{4}$	$2\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{7}{8}$
$1\frac{3}{4}$	$1\frac{25}{32}$	$\frac{7}{16}$ x $\frac{3}{32}$	2	$\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{7}{8}$
2	$2\frac{1}{32}$	$\frac{1}{2}$ x $\frac{1}{8}$	$2\frac{3}{8}$	$\frac{3}{8}$	3	$\frac{3}{8}$	$1\frac{3}{8}$	1
$2\frac{1}{4}$	$2\frac{9}{32}$	$\frac{1}{2}$ x $\frac{1}{8}$	$2\frac{5}{8}$	$\frac{3}{8}$	$3\frac{1}{4}$	$\frac{3}{8}$	$1\frac{3}{8}$	1
$2\frac{1}{2}$	$2\frac{17}{32}$	$\frac{5}{8}$ x $\frac{5}{32}$	$2\frac{7}{8}$	$\frac{3}{8}$	$3\frac{3}{4}$	$\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$
$2\frac{3}{4}$	$2\frac{25}{32}$	$\frac{5}{8}$ x $\frac{5}{32}$	$3\frac{1}{8}$	$\frac{3}{8}$	4	$\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$
3	$3\frac{1}{32}$	$\frac{3}{4}$ x $\frac{3}{16}$	$3\frac{1}{2}$	$\frac{1}{2}$	5	$\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{3}{8}$
$3\frac{1}{4}$	$3\frac{9}{32}$	$\frac{3}{4}$ x $\frac{3}{16}$	$3\frac{3}{4}$	$\frac{1}{2}$	5	$\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{3}{8}$
$3\frac{1}{2}$	$3\frac{17}{32}$	$\frac{7}{8}$ x $\frac{7}{32}$	4	$\frac{1}{2}$	6	$\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$
$3\frac{3}{4}$	$3\frac{25}{32}$	$\frac{7}{8}$ x $\frac{7}{32}$	$4\frac{1}{4}$	$\frac{1}{2}$	6	$\frac{5}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$

Note.—Use pins with Lomas nuts in preference to cotter pins whenever possible.

TABLE LXXVII.

EYEBARS.*

(AMERICAN BRIDGE CO.)



Width of Bar.	Min. Thickness of Bar.	Head.			Screw End.			Min. Thickness of Bar.
		Diam.	Max. Pin.	Additional Mat. for Head.	Additional Mat. for Upset.	Diam.	Length.	
3	$\frac{3}{4}$"	7"	3"	1' 3"	1' 5"	$2\frac{1}{2}$"	$5\frac{1}{2}$"	1"
	$\frac{3}{4}$	8	$3\frac{3}{4}$	1 6	1 5	$2\frac{3}{4}$	6	$1\frac{1}{8}$
4	$\frac{3}{4}$	$9\frac{1}{2}$	4	1 8	1 8	3	6	1
	$\frac{3}{4}$	$10\frac{1}{2}$	5	1 10	1 8	$3\frac{1}{4}$	$6\frac{1}{2}$	$1\frac{3}{16}$
5	$\frac{3}{4}$	$11\frac{1}{2}$	$4\frac{3}{4}$	1 9	1 9	$3\frac{1}{4}$	$6\frac{1}{2}$	1
	1	$12\frac{1}{2}$	$5\frac{3}{4}$	2 1	1 9	$3\frac{1}{2}$	7	$1\frac{1}{8}$
6	$\frac{3}{4}$	$13\frac{1}{2}$	$5\frac{1}{2}$	1 11	1 11	$3\frac{3}{4}$	8	$1\frac{1}{8}$
	1	$14\frac{1}{2}$	$6\frac{1}{4}$	2 2	1 11	4	8	$1\frac{1}{4}$
7	$\frac{7}{8}$	16	$6\frac{3}{4}$	2 3	2 3	$4\frac{1}{4}$	9	$1\frac{1}{4}$
	$\frac{15}{16}$	17	$7\frac{1}{2}$	2 8	2 3	$4\frac{1}{2}$	9	$1\frac{3}{8}$
	1	17	6	2 3				
8	$1\frac{1}{16}$	18	7	2 6				
	$1\frac{1}{8}$	$18\frac{1}{2}$	$7\frac{1}{2}$	2 10				

* NOTE.—Eye-bars are hydraulic forged, and will develop the full strength of the bar, under conditions given in the above table, when tested to destruction. The maximum sizes of pins given in the above table allow an excess in sectional area of head on lines "SS" over that of the body of the bar of 33 per cent. for diameter of pins, not larger than the width of the bar and 36 per cent. for pins of larger diameter than the width of the bar.

"Full size test of steel eye-bars shall be required to show not less than 10 per cent. elongation in the body of the bar, and tensile strength not more than 5,000 pounds below the minimum tensile strength required in specimen tests of the grade of steel from which they are rolled. The bars will be required to break in the body, but should a bar break in the head, but develop 10 per cent. elongation and the ultimate strength specified, it shall not be cause for rejection, provided not more than one third of the total number of bars tested break in the head; otherwise the entire lot will be rejected.

"The heads of eye-bars shall not be less in strength than the body of the bar.

"The heads of eye-bars shall be made by upsetting, rolling, or forging into shape. Welds in the body of the bar will not be allowed.

"The bars must be perfectly straight before boring.

"The holes shall be in the centre of the head and on the centre line of the bar.

"All eye-bars shall be annealed.

"Bars which are to be placed side by side in the structure shall be bored at the same temperature, and shall be of such equal length that, upon being piled on each other, the pins shall pass through the holes at both ends at the same time without driving."

APPENDIX.

PAGE 20. In many shops, a custom — and a good one — obtains of covering the exposed parts of shafts of twin-screw steamers with ratline laid in paint, when the shafts are not cased with brass.

The use of pins or tap-rivets as an aid in holding shaft casings does not meet with universal approval, the argument against them being that, if the shrinkage of the casing does not fully secure the latter, no pins will; and, further, that often the putting in of the pins tends to loosen the casing.

PAGE 34. The expression,

$$\frac{R_1^2 - R_0^2}{2R_1^2} \cdot \theta_0$$

is simply the value of P_1 obtained from the third equation of (27) by making P_0, in that equation, equal to zero.

PAGE 260. The "Flat Key" is the type used almost exclusively in marine work.

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